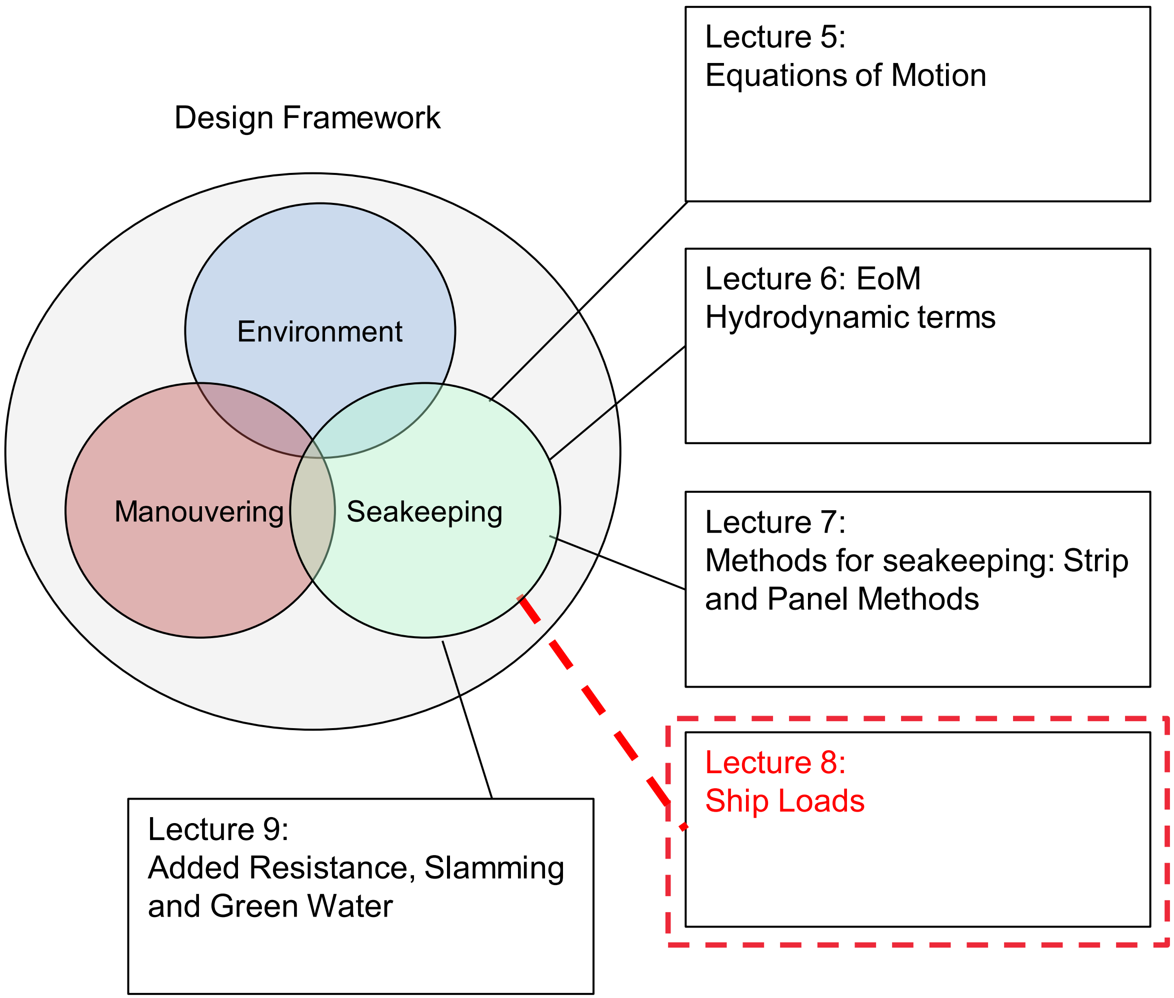
Aalto University
School of Engineering

MEC-E2004 Ship Dynamics

Lecture 8 – Ship Loads

Aalto University
School of Engineering

Where is this lecture on the course?



Lecture 5:
Equations of Motion

Lecture 6: EoM
Hydrodynamic terms

Lecture 7:
Methods for seakeeping: Strip and Panel Methods

Lecture 8:
Ship Loads

Lecture 9:
Added Resistance, Slamming and Green Water

Contents

Aim: The aim is to understand practical issues on globally wave induced loads and ways of computation. The focus is primarily on linear computations and assessment of results based on RAOs:

- Outline of key problems and practical implications
- Linear approach and a note on non-linear effects
- Brief Introduction to Hydroelasticity
- Calculation of the spectral parameters and properties
- Short term load predictions

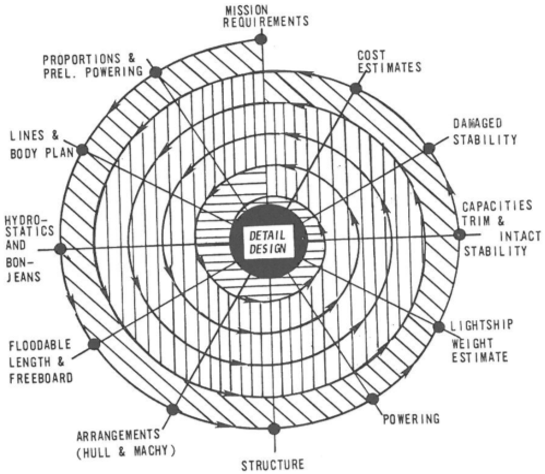


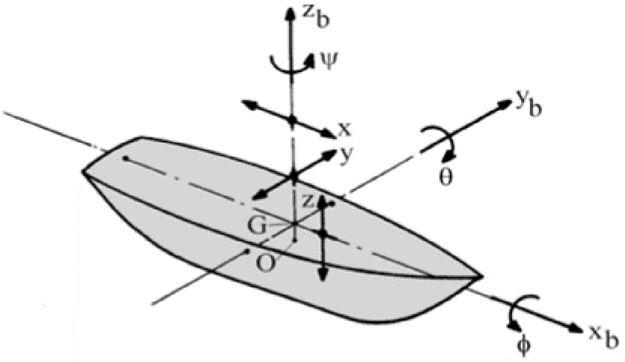
Fig. 1 Basic design spiral

Literature:

- Lewis, "Principles of Naval Architecture – Vol. III", SNAME, 1989
- Yong Bai, Chapter 2 - Wave Loads for Ship Design and Classification, In Marine Structural Design, Elsevier Science, Oxford, 2003, Pages 19-37, ISBN 9780080439211, https://doi.org/10.1016/B978-008043921-1/50002-2.
- Bishop RED and Price WG, Hydroelasticity of Ships, Cambridge University Press ISBN 0521223288
- Kukkanen, T. Spectral Fatigue Analysis for Ship Structures. Uncertainties in Fatigue Actions. TKK, Konetekniikka, Lis.työ. 1996.

Assignment 4

- Grades 1-3:
  - Select a book-chapter related to determination of ship motions and loads and get acquainted with a tool to predict these
  - Form a seakeeping analysis model from your ship, discuss the simplifications made
  - Perform the computations for Response Amplitude Operators
- Grades 4-5:
  - Compute all motions (6) and global loads (bending moments and shear forces) for your ship for selected sea spectra (e.g. worst case spectra in North Atlantic). You can predict 3 hour maximums
  - Based on scientific literature, discuss the accuracy of your results



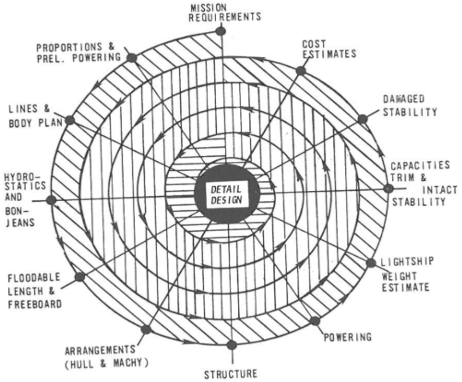
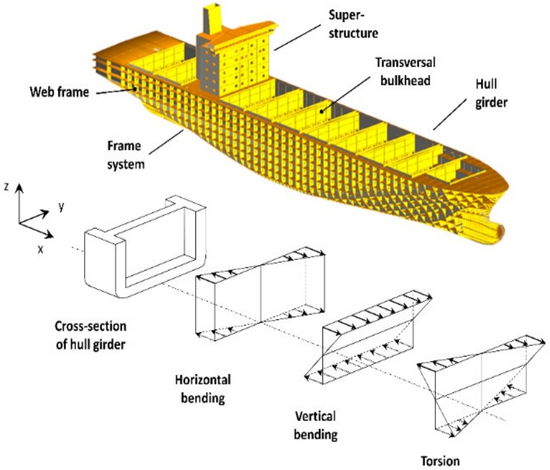


Fig. 1 Basic design spiral

Motivation – Hull Girder Loads

- In the structural design of the ships, a common practice is to express the design loads by sagging and hogging bending moments and shear forces
- The sagging and hogging bending moments and shear forces are hull girder loads
- The hull girder loads are balanced by internal forces and moments affecting the cross-section of the ship hull (stress resultants)
- The accurate prediction of the extreme wave loads is important for safety
- For ships in a heavy seas, the sagging loads are larger than the hogging loads
- Linear theories cannot predict differences between sagging / hogging loads



Motivation – Hull Girder Loads


The Erika Disaster

Who said ships do not fail
In still water conditions or at Harbour?

273-foot barge ITB-270 With Gravel Was Being Loaded Along The Duwamish River At The Ash Grove Cement Co. Terminal, Harbor Island, 4pm on May 2 2008. ITB-270 Suddenly Broke In Half. One Crew Aboard Was Rescued. ITB-270 Settled In Shallow Water Off Harbor Island.

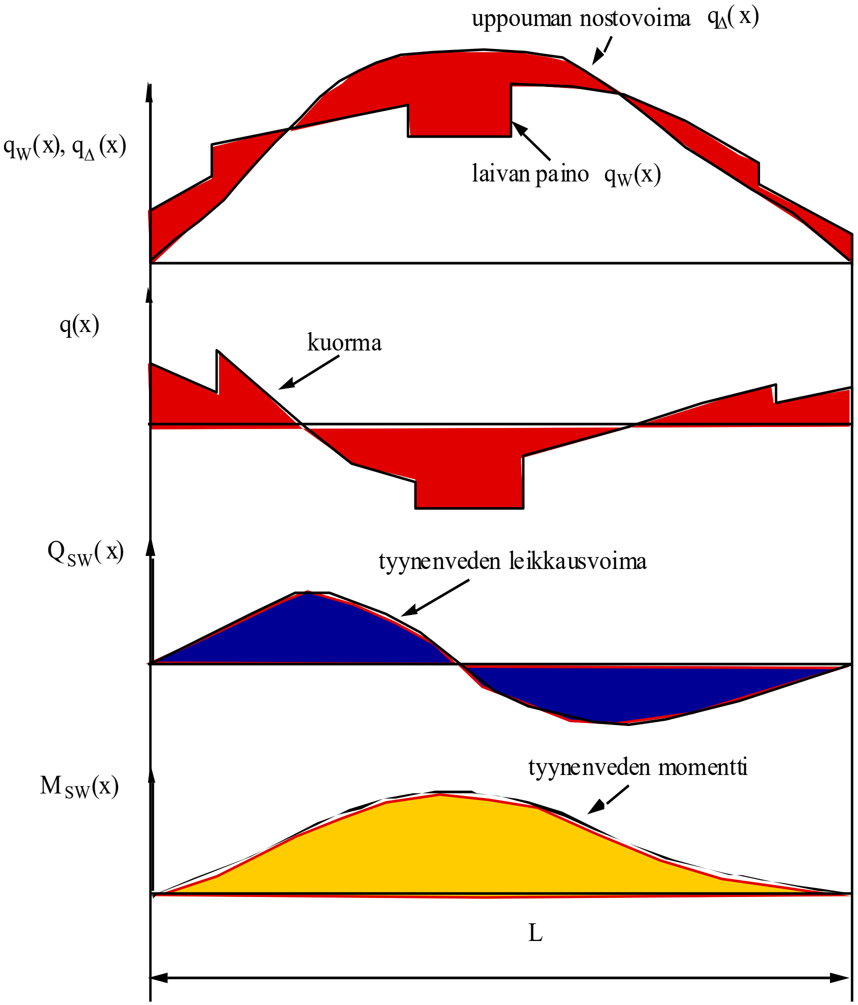
"It's always concerning when a vessel buckles like that, or a barge buckles like that. It's not supposed to happen."
Petty Officer Michael Donaldson - U.S. Coast Guard

Mass vs hull/water interaction forces

- Loads relate with the hydrodynamic actions. These forces relate with:
    - ✓ Mass distribution
    - ✓ Hull/water interactions (hydro-static/dynamic)

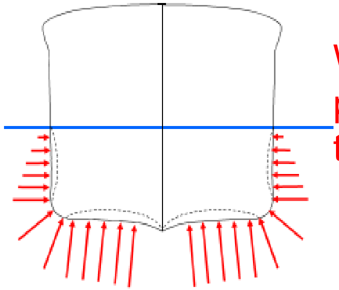
- In waves the vessel experiences accelerations due to ship motions. As a result the inertia component is added to the weight.

- The pressure acting on a hull surface in waves comprises of
    - ✓ hydrostatic;
    - ✓ radiation;
    - ✓ Froude-Krylov
    - ✓ Diffraction contributions

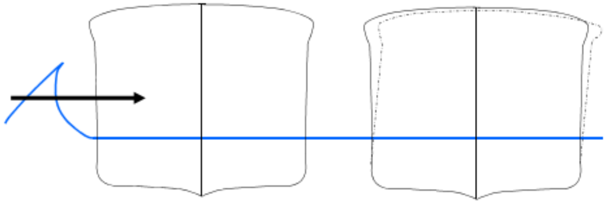


Global Loads

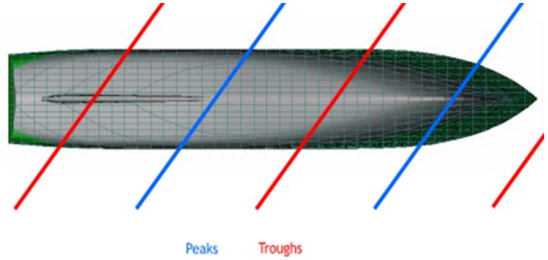
Global Loads caused by wave actions may be divided into (1) hydrostatic pressure, (2) racking, (3) torsion, (4) hogging/sagging due to waves, (5) still water hogging/sagging.



Water or hydrostatic pressure increases in a linear relationship with depth. This pressure of water tries to crush the ship and so the structure should be designed to resist such forces



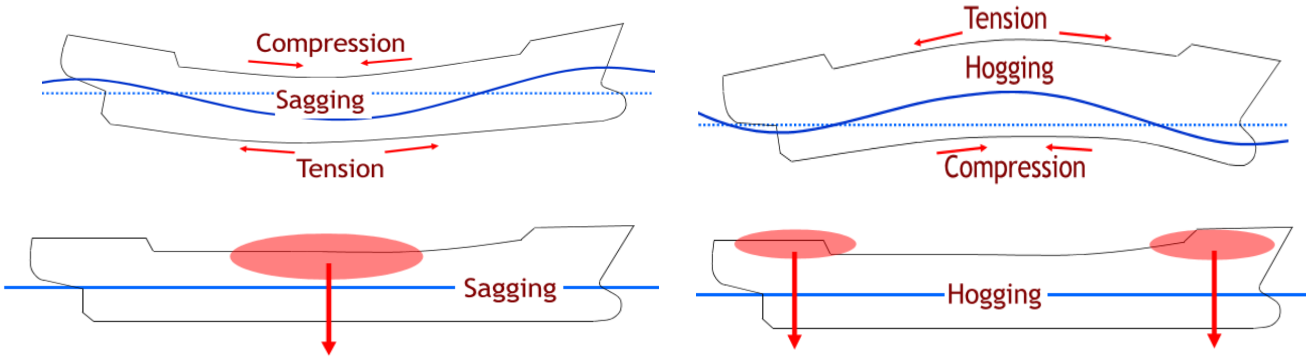
Transverse loads from waves (or tugs) may cause the vessel to distort sideways



Torsion loads are twisting loads along the hull caused by quartering seas. This raises loading in way of deck edges and bilge turn

Global Loads – Hogging and Sagging

- **Hogging and sagging** are the major global loads the ship must survive. They create tension and compression in way of the keel/deck. **In still water** – in the lightship condition – the mass and buoyancy forces are equal and opposite and generally act fairly evenly along the ship. **In waves** the distribution of buoyancy changes especially in head seas and following seas. This is most apparent where the wave length is similar to ship length.

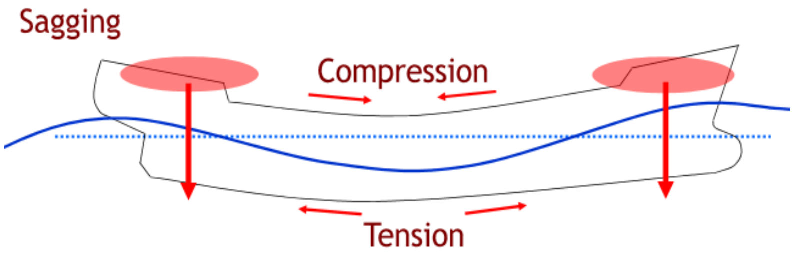


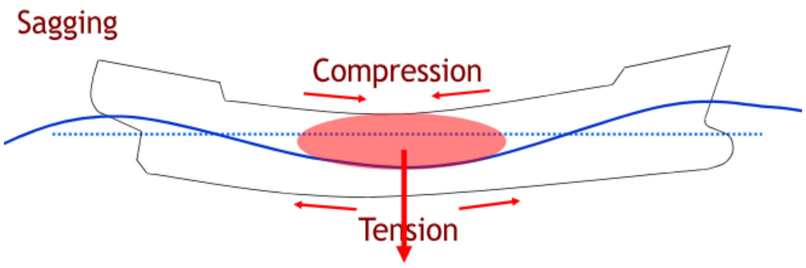
- **Sagging :** *Peaks are at bow/stern and the trough amidships.* The buoyancy force shifts to the ends of the vessel. The bow experiences forces upwards by buoyancy and the vessel in pulled downwards amidships by gravity.

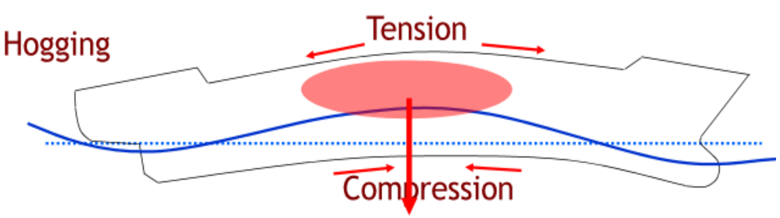
- **Hogging:** *Troughs at bow/stern and peaks amidships.* The buoyancy force is shifted to the centre of the vessel. Amidships is forced upwards by buoyancy and the bows are pulled downwards by gravity.

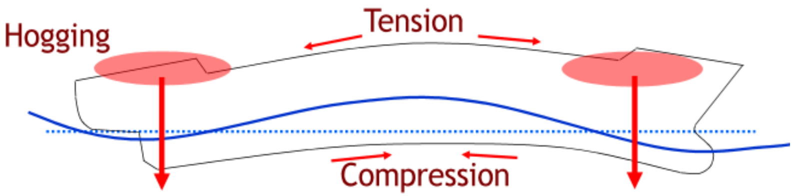
Global Loads – Hogging and Sagging

When hogging / sagging due to cargo distribution (still water) and waves are added up the hull girder may experience significant stresses. Cargo loading and ship navigation are critical to avoid exceeding such stress levels.







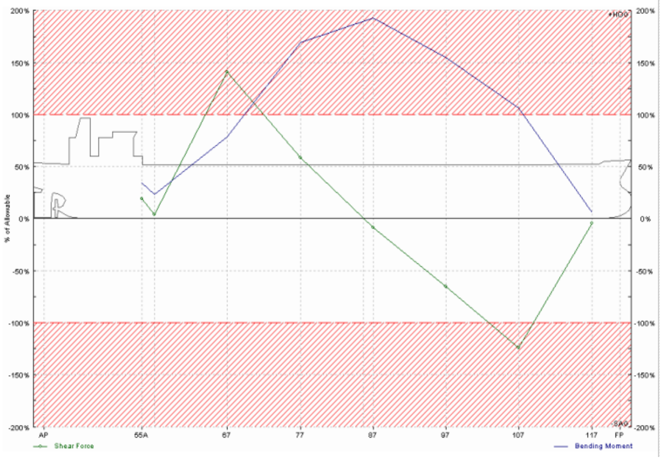


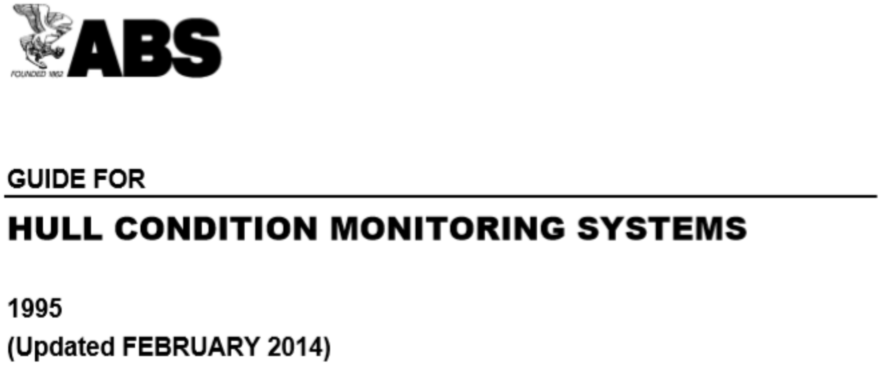
Compression on deck and tension in bottom is called a sagging condition

Tension in deck and compression in a bottom is called hogging condition

Loading Instruments and HCM (Sensors)

Loading cargo gives rise to loads such as BM and SF. These values are limited by still water limits that should not be exceeded during loading operations. They usually include a big safety margin that accounts for the effects of waves in open sea conditions and loading operations at port. A loading instrument (or loading computer) onboard helps to check loads during loading operations or at open seas. Hull condition monitoring systems are also used.

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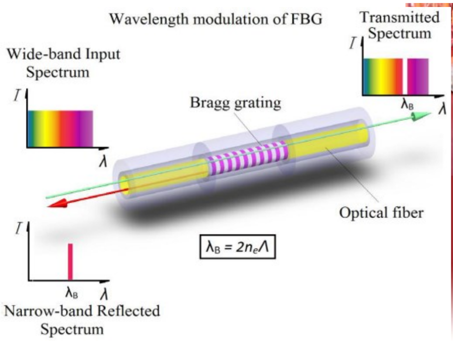
FOUNDED 1862

GUIDE FOR

HULL CONDITION MONITORING SYSTEMS

1995

(Updated FEBRUARY 2014)



Long Base Strain Gauge

Short Base Strain Gauge

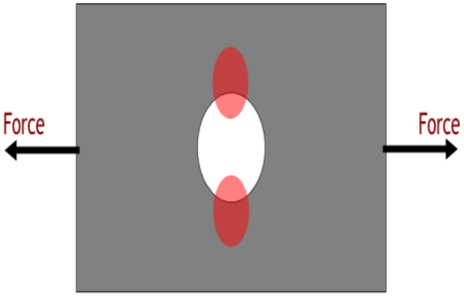
Wei Shen, Renjun Yan, Lin Xu, Guangning Tang, Xialiang Chen, Application study on FBG sensor applied to hull structural health monitoring, Optik - International Journal for Light and Electron Optics, Volume 126 (17)- 2015 ; https://doi.org/10.1016/j.ijleo.2015.04.046.

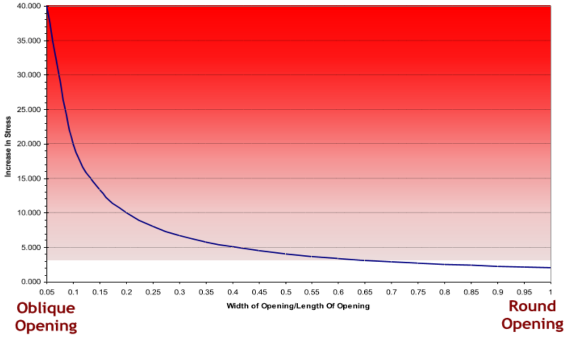
From Loads to stresses

- The wave-induced primary stresses are important for:
    - The dynamic response in waves
    - The ultimate strength assessment of the hull girder and plates
    - The fatigue strength analysis of structural details

- Stresses may amplify by stress concertation points known as discontinuities

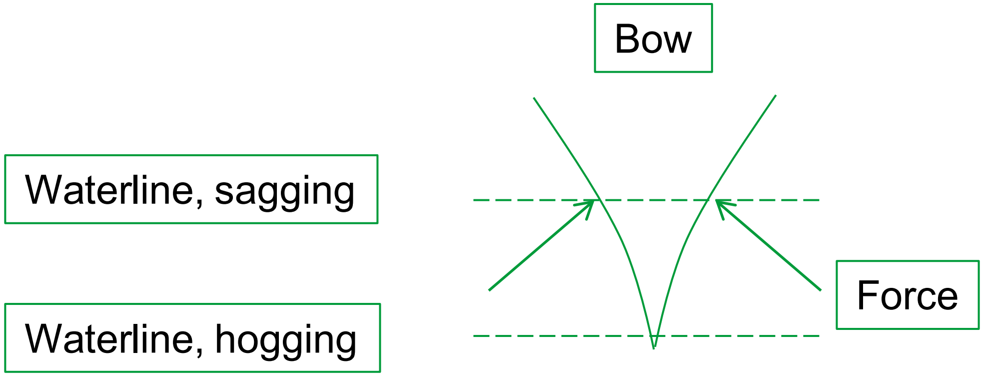
- Hull Condition monitoring systems can help us to monitor these stresses as well as BM and SF in service
    - The dynamic response in waves
    - The ultimate strength assessment of the hull girder and plates
    - The fatigue strength analysis of structural details





Wave crests at bow/stern (different prespective)

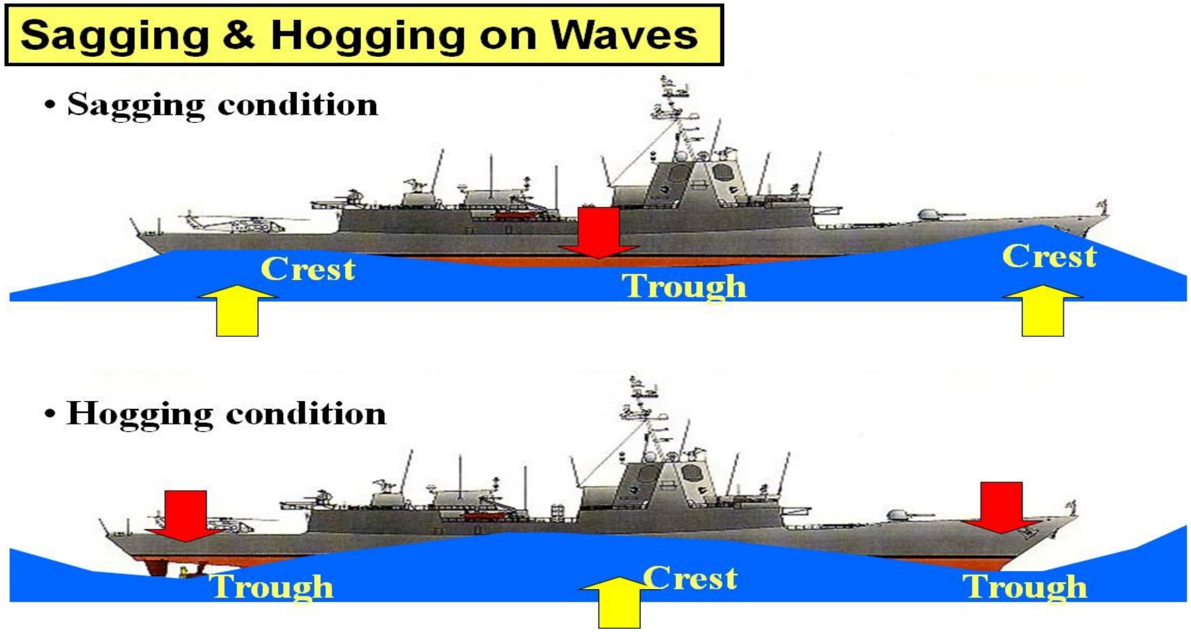
- In waves the sagging condition occurs if wave crests are at the bow and stern and hogging if a wave crest is at mid-ship
- The sagging increases if the ship has large bow flare and the ship motions are large with respect to waves
- The stern form of the ship can have the same effect if the ship has a flat bottom stern close to the waterline.


Bow

Waterline, sagging

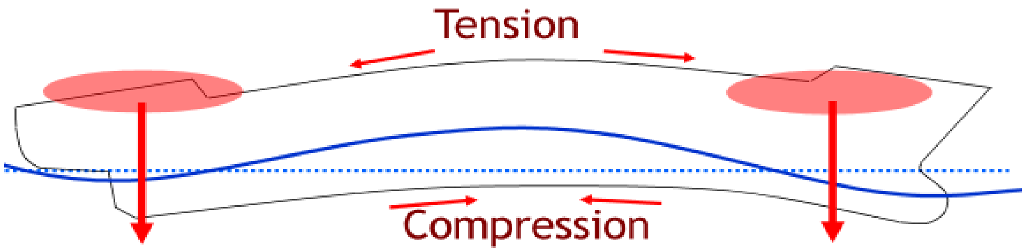
Waterline, hogging

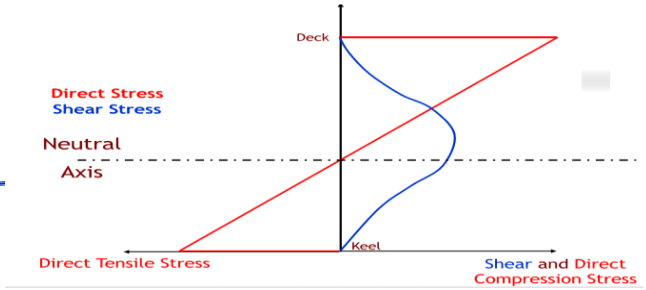
Force

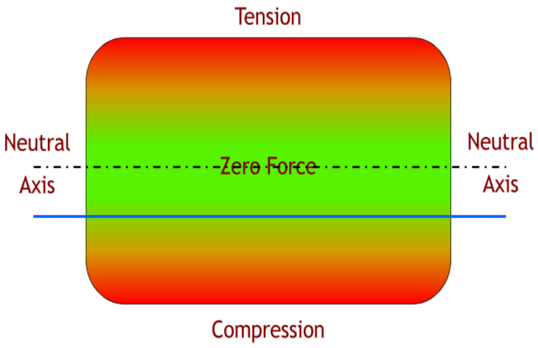


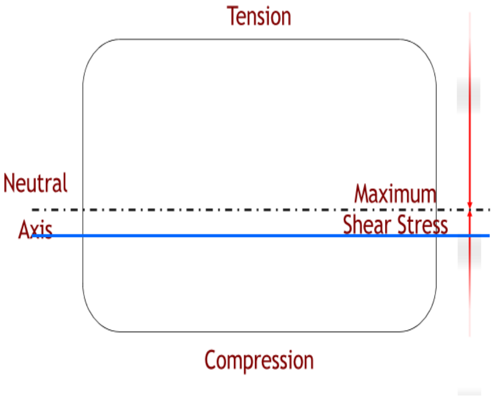
From Gobal Loads to Stresses

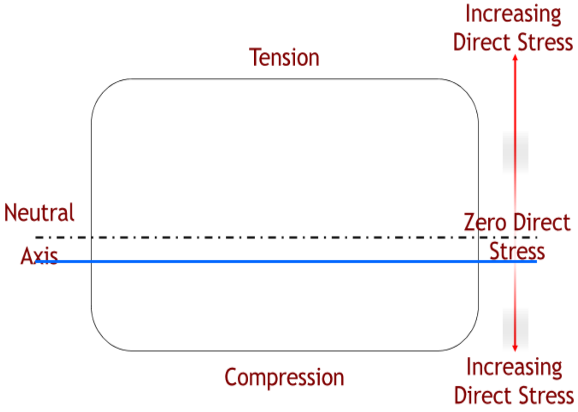
Stresses generated by hogging / sagging are absorbed by the main longitudinal items of the ship in way of the deck and the keel. Stresses vary along the hull depth.











Quasi Static Response - basics

- In Quasi static analysis we use simple beam theory and assume :
  - Loads and deflections have a single value at any cross section
  - The hull girder remains elastic with small deflections and strain due to bending varies linearly
  - In way of the neutral axis
  - Static equilibrium applies

Static equilibrium

Total buoyancy force = ship weight

LCB = LCG.

$$\rho g \int_0^L a(x)\,dx = g\int_0^L m(x)\,dx = g\Delta$$

where :

$a(x)$ = immersed cross-sectional area

$m(x)$ = mass distribution

ρ = density of seawater

g = gravitational acceleration

Δ = displacement

Dynamic Equilibrium

$$\rho g \int_0^L a(x)\,x\,dx = g\int_0^L m(x)\,x dx = g\Delta l_G$$

where :

l_G = distance from origin to l.c.g

Quasi Static Response – weights & beam theory

- The weight will not equal the buoyancy at each location along the ship. The weights are the combination of lightship and cargo weights (more or less fixed).

- The buoyancy forces are determined by the shape of the hull and the position of the ship in water (draft & trim). The net buoyancy will adjust itself until it exactly counteracts the net weight force.

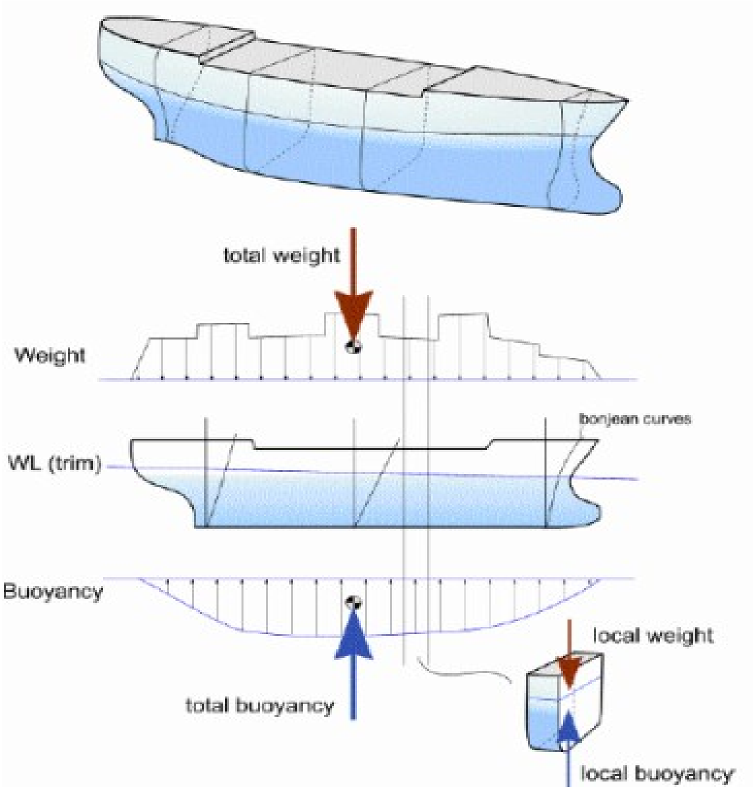
- Local segments may have more or less weight than the local buoyancy. The difference will be made up by a transfer of shear forces along the vessel.

- The governing equation for BM is $\frac{d^2M}{dx^2} = f(x)$

where f(x) represents the loading of a ship as a beam

- The net distributed force is given by the resultant between weight and buoyancy forces

$$f(x) = b(x) - w(x)$$



Quasi Static Response – weights & beam theory

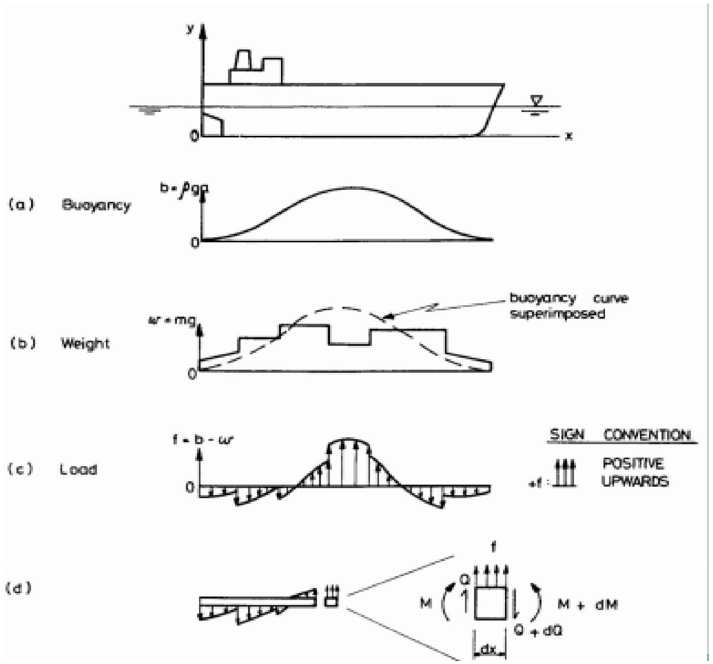
- To solve for M(x) we need to know transverse Shear force Q(x). Summing the moments over a differential element gives :

$$Q(x)=\int_0^x f(x)dx$$

$$M(x)=\int_0^x Q(x)\,dx$$

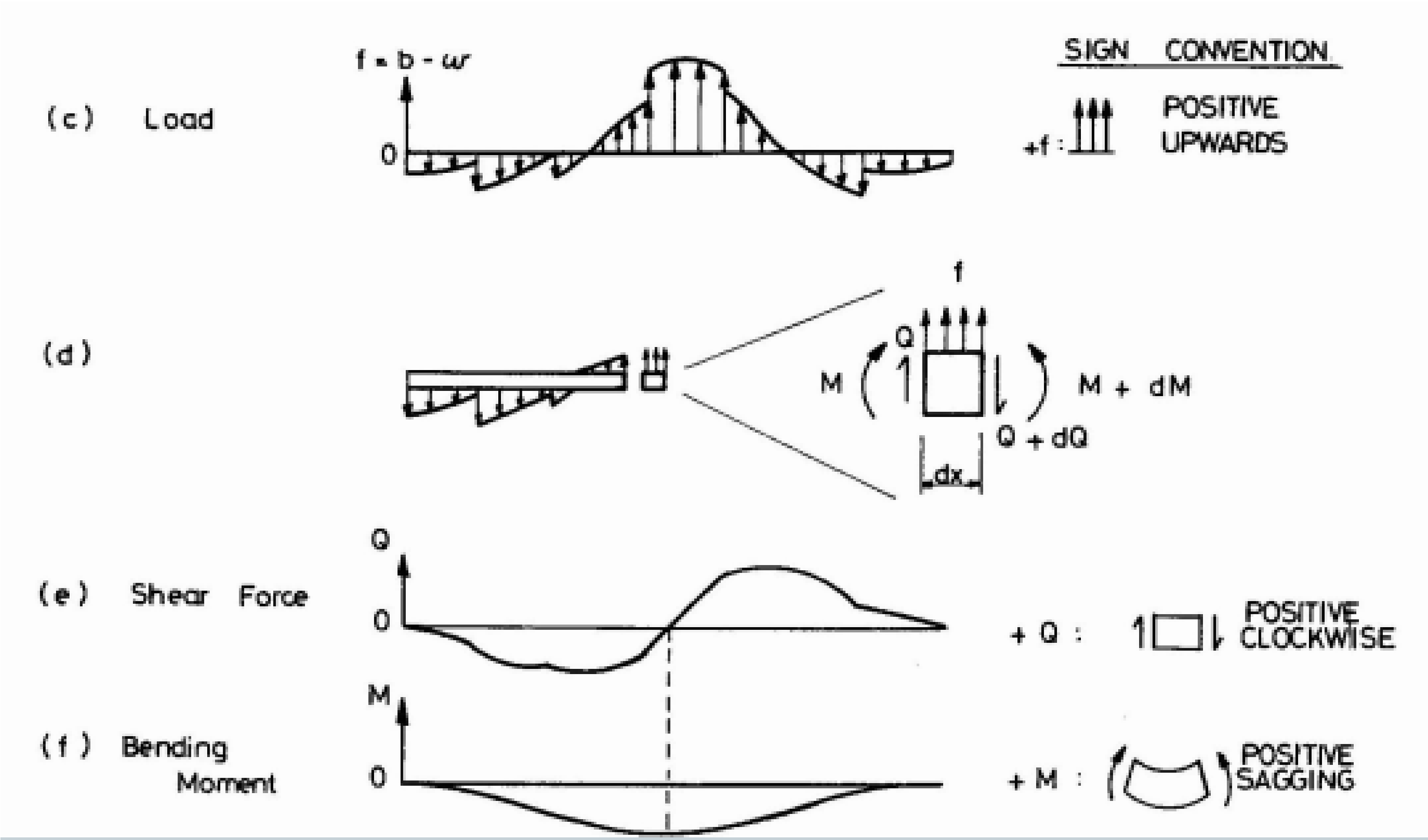
- <u>Conventions :</u> +ve shear → clockwise rotation; +ve BM → concave upwards (or sagging); -ve BM concave downwards (hogging).

- Two buoyancy forces to consider (a) **still water buoyancy** which is a static quantity given as the function of the hull shape; (b) **wave buoyancy** which is dynamic/probabilistic quantity.



- The buoyancy distribution in waves is calculated separately an superimposed on SW buoyancy force. The SW buoyancy distribution is determined from static and moment equilibrium equations. So we need to know the mass distribution m(x) or at least the displacement / location of LCG.

Quasi Static Response – weights & beam theory



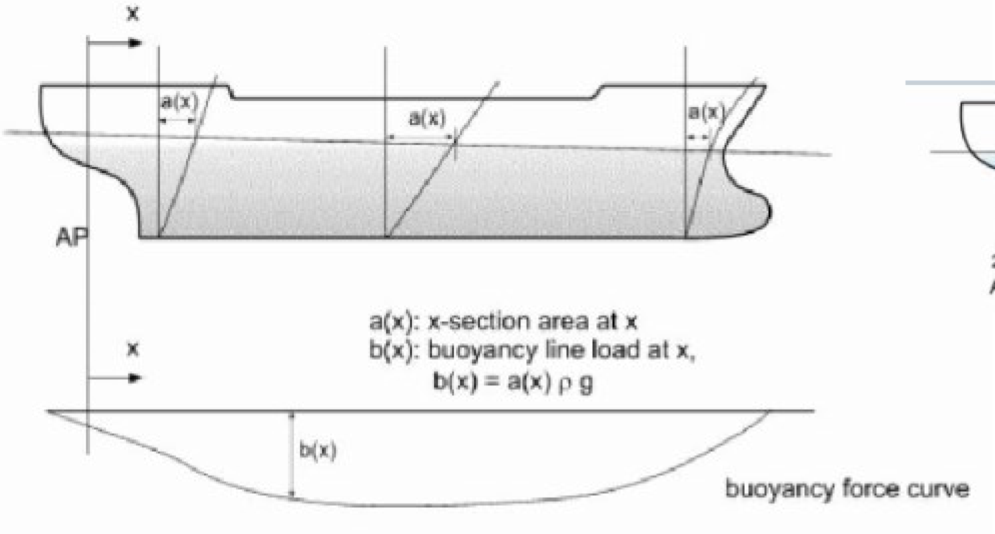
- Load = zero → Max/min SF/BM
- In general SF = 0 amidships / bow / stern and peaks near 1/4 points
- In general BM = max near amidships and BM = 0 at bow/stern

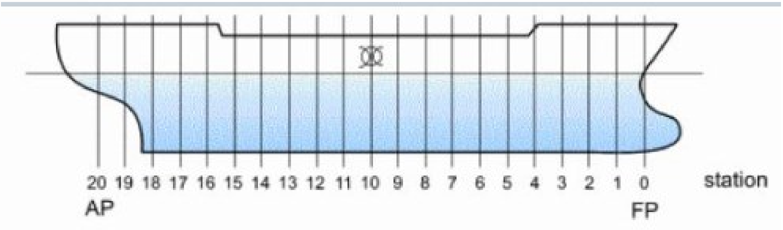
Quasi Static Response – Bonjean Curves

- The local buoyancy / meter can be determined form the cross – sectional area of the hull at discrete locations. This area depends on local draft and it is found using the bonjean curves.

- Each bonjean curve corresponds to each station. **If there are for example 21 stations form FP to AP we can divide the LBP in 20 segments.**

Hull Form Bonjean + (Draft + Trim) → Buoyancy





Quasi Static Response – Bonjean Curves

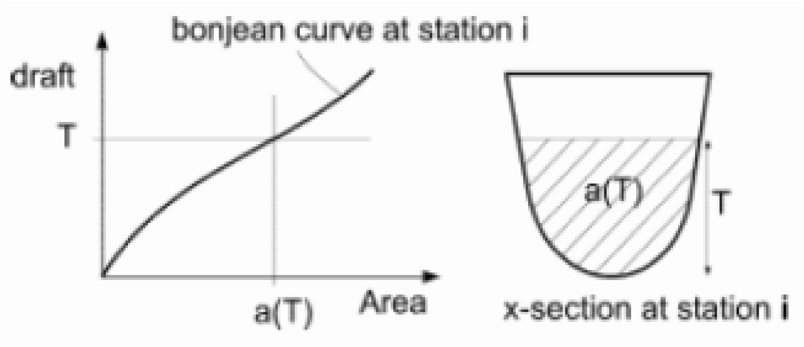
- At each station a curve of the cross sectional area is drawn. Bonjean curves are shown on the profile of the vessel and we use them to determine the buoyancy distribution at the waterline.

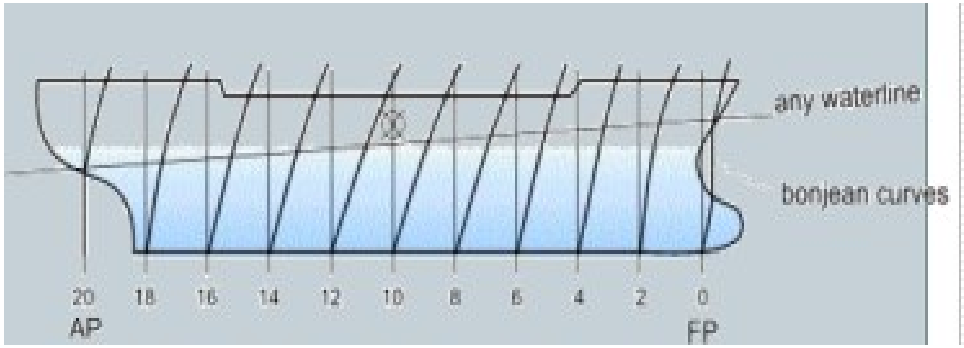
- The total displacement at a given draft/ trim is found by summing up contribution of each segment.

$$\nabla = \sum_{i=0}^{20} \left\{ a_i(T_i) \cdot \frac{LBP}{20} \right\} [\mathrm{m}^3]$$

- The line load Is then given at each station as

$$\Delta_i = \nabla_i \cdot \rho \cdot g$$



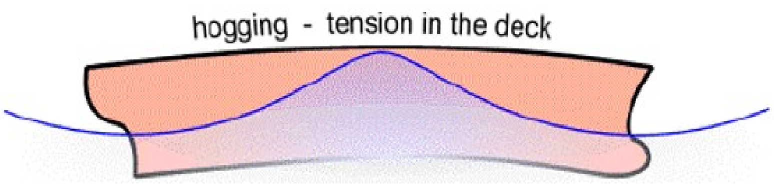


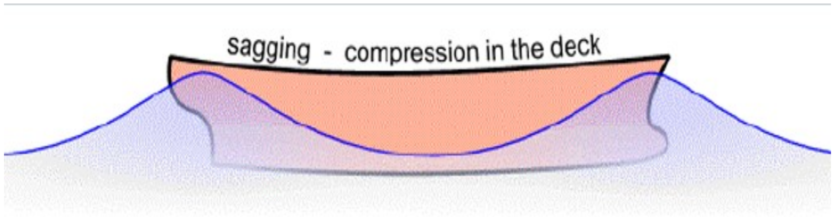
Quasi Static Response – wave actions

Two basic RULES : The 1/20 Rule and the L = lambda Rule

- When we consider the wave forces on the ship to be quasi static it means that they can be treated as a succession of equilibrium states.

- **MAX hogging BM** occurs when the ships' mid body is on the crest of the wave . Conversely, **MAX sagging BM** occurs when the mid-body is on the trough and the bow / stern are on crests.





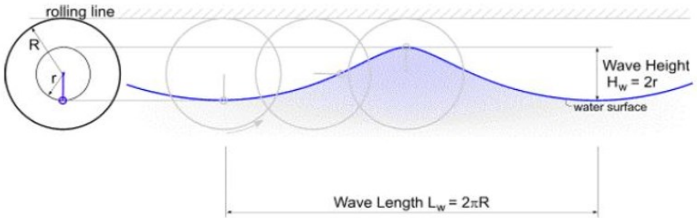
L = lambda RULE

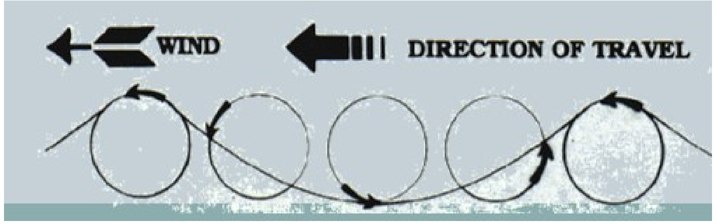
The highest BM will occur when the wavelength approaches the vessel length. The design wave of a vessel therefore has a wavelength equal to the vessel length

Quasi Static Response – wave actions

- The shape of an ocean wave is often depicted as a sine wave, but waves at sea can be better described as **"trochoidal".** A trochoid can be defined as the curve traced out by a point on a circle as the circle is rolled along a line. The discovery of the trochoidal shape came from the observation that particles in the water would execute a circular motion as a wave passed without significant net advance in their position.

- The motion of the water is forward as the peak of the wave passes, but backward as the trough of the wave passes, arriving again at the same position when the next peak arrives. (Actually, experiments show a slight advance of the water with the waves, but that advance is small compared to the overall circular motion.)





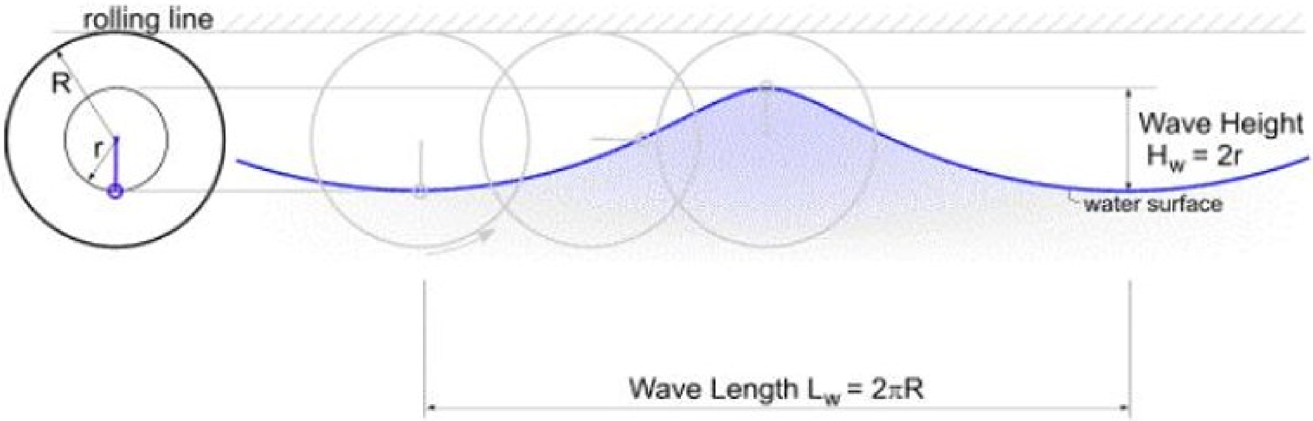
1/20 RULE

The wave height (peak to through) may be generally assumed to be the 1/20th of the wave length else the ship will break (1/20 RULE).

Quasi Static Response – wave actions

- For trochoidal waves Lw = LBP , Hw=Lbp/20 → Lw = 2*pi*R and Hw = 2*r
- This gives : $R = \frac{L_{BP}}{2\pi}, r = \frac{L_{BP}}{40}$ and, $\frac{r}{R} = \frac{\pi}{20}$
- So the wave shape is defines as :

$$\boxed{\begin{aligned} x &= R\theta - r\sin\theta \\ z &= r(1-\cos\theta) \end{aligned}} \Rightarrow \boxed{\begin{aligned} x &= \frac{L}{2\pi}\theta - \frac{L}{40}\sin\theta \\ z &= \frac{L}{40}(1-\cos\theta) \end{aligned}}$$



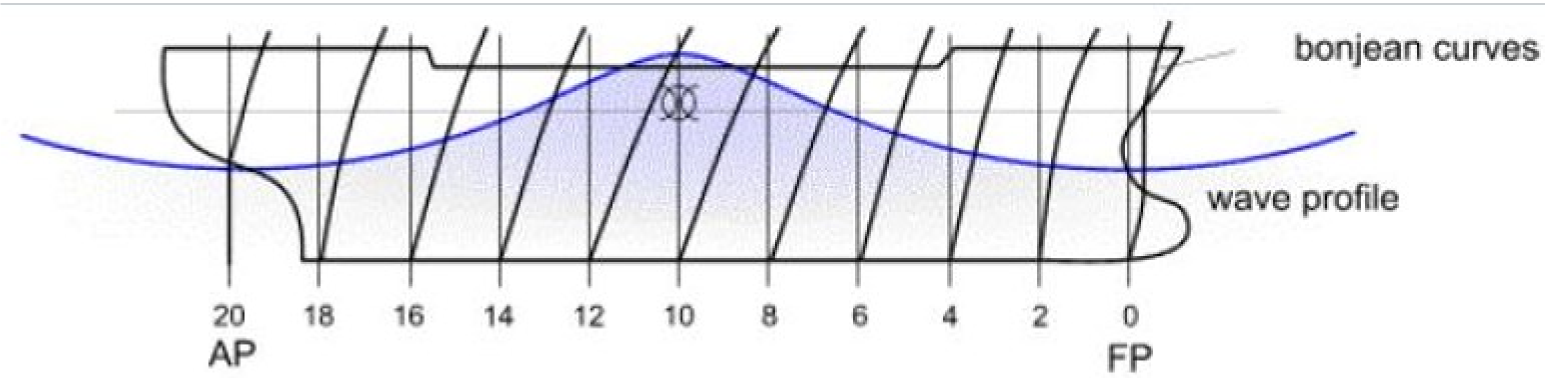
Quasi Static Response – wave actions

- The L/20 rule for wave height has been shown to be overly conservative for large vessels and a more modern formula is:

$$H_W = 0.607\sqrt{L_{BP}} \text{ (in metres)} \qquad R = \frac{L_{BP}}{2\pi}, r = 0.303\sqrt{L_{BP}}$$

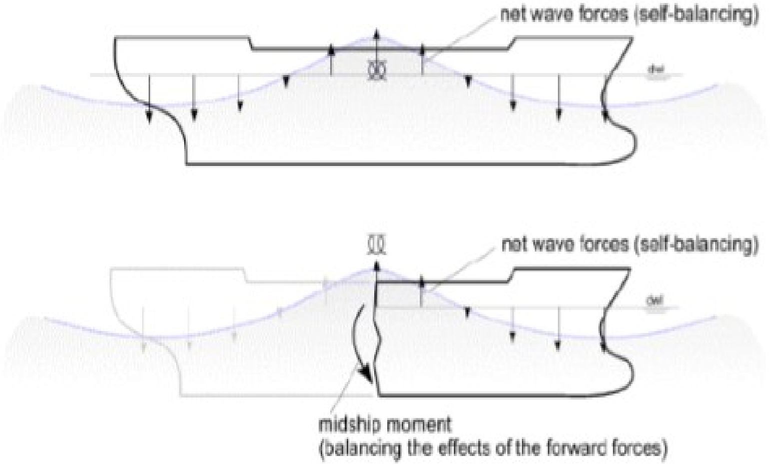
- Hughes suggests that for ships of length greater than 350 m : $H_W = \dfrac{227}{\sqrt{L_{BP}}}$ (in metres)

<u>Question :</u> How can we calculate the wave bending moments by placing the ship on the design wave and using the Bonjean curves ?



Quasi Static Response – process

1. Obtain Bonjean curves
2. At each station determine the still water buoyant forces (using the design draft)
3. At each station determine the total buoyancy forces using the local draft in that part of the wave
4. The net wave buovancy forces are the difference between the total and still water buoyancy forces $F_{i,wave} = F_{i,w_t} - F_{i,SW}$
5. From here we have a set of buoyancy forces due to waves, which are in equilibrium
6. We calculate the BM amidships from the net effect of forces either fore or aft



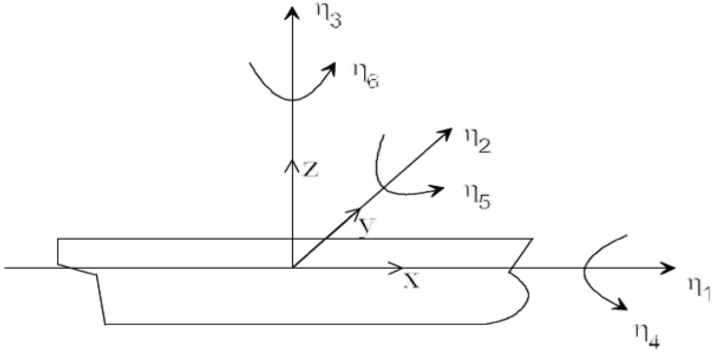
We can use computer packages to find the BM

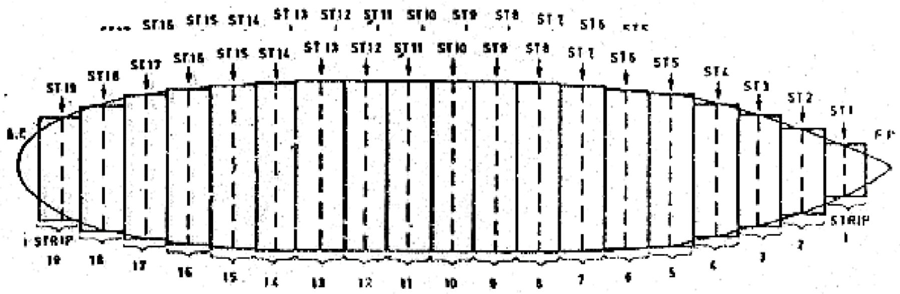
Using a hull model, the buoyant forces on the fore and aft ends of the hull can be determined by the volume and centroid of the submerged volumes at a specific waterline surface. A similar procedure could be used to determine the wave values, but the waterline surface would be the trochoidal wave profile.

What about strip theory?

Linear approach to wave loads

- Assume the body-fixed co-ordinate system, which is often called ‘the seakeeping co-ordinate system’
- For simplicity we shall limit the discussion to global VSF and VBM acting on a rigid hull



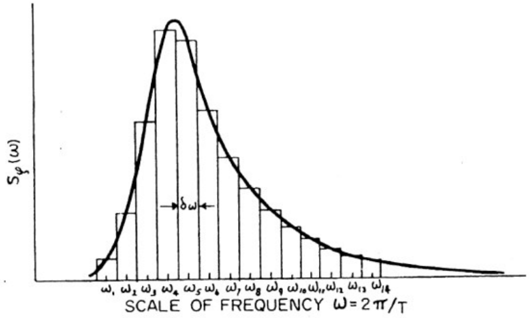


$$\sum_{k=1}^{6}\left[\left(M_{jk}+A_{jk}\right)\ddot{\eta}_k+B_{jk}\dot{\eta}_k+C_{jk}\eta_k\right]=F_j e^{-i\omega t},\ j=1,2,\ldots,6$$

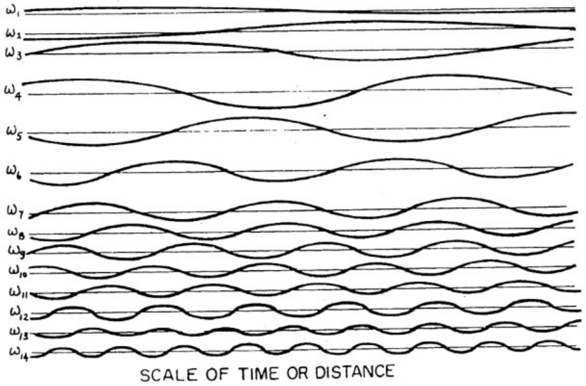
$$\left|H(\omega)\right|^2=\left[\frac{Y(\omega)}{A(\omega)}\right]^2$$

RAOs

- The model is linear.
- We can use the concept of RAOs to relate the loads to the wave and ship operating conditions (wave length, heading and ship speed)
- That is we can proceed similarly as we did with the other linear responses and derive a short term internal load prediction for a ship operating in irregular waves.
- One shortcoming of the linearity assumption is that the result does not distinguish between the sagging and the hogging condition except for the still water condition.
- Other short comings maybe related with forward speed effects, wave elevation in way of the free surface, large amplitude motions etc.



(a) SPECTRUM



(b) COMPONENT WAVES

Fig. 8 Typical variance spectrum of waves, showing approximation by a finite sum of components

Hirdaris, S.E. and Lee, Y. and Mortola, G. and Incecik, A. and Turan, O. and Hong, S.Y. and Kim, B.W. and Kim, K.H. and Bennett, S. and Miao, S.H. and Temarel, P. (2016) The influence of nonlinearities on the symmetric hydrodynamic response of a 10,000 TEU container ship. Ocean Engineering, 111. 166–178. ISSN 0029-8018

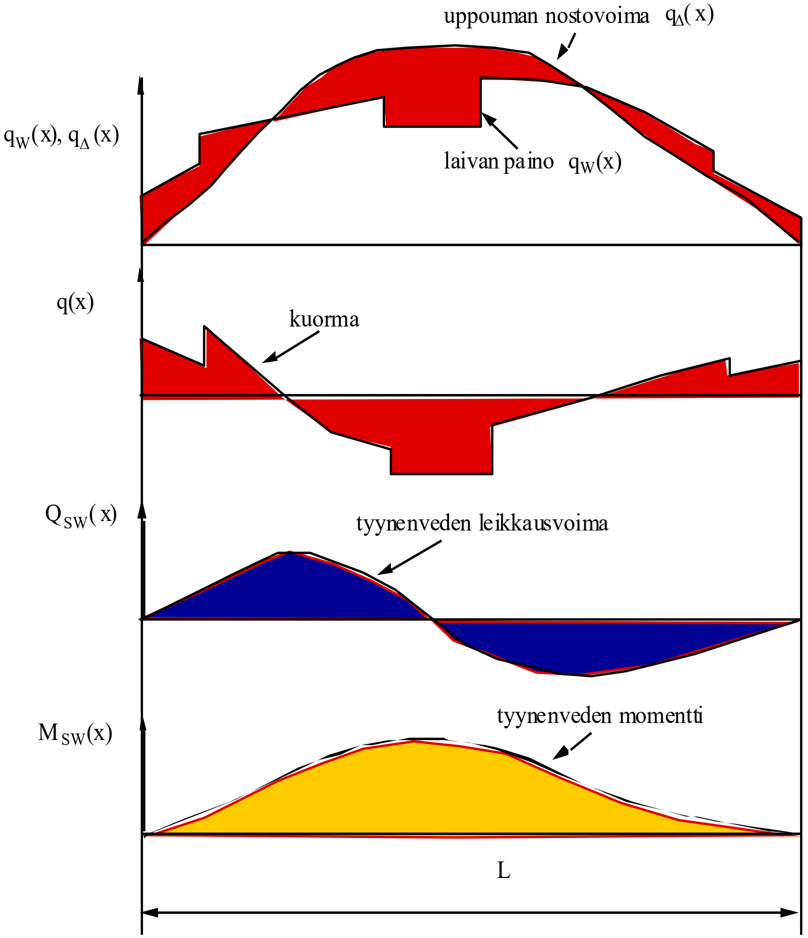
The still water condition

- At each station, denoted by a position *x*, we have the vertical force per unit length given by a sum of weight and buoyancy at this section that is:

$$q(x) = -m(x)g + \rho g A(x)$$

- **If the ship is heaving and pitching we have consider inertia and hydrodynamic *F(x)* loads.**
- **The vertical force per unit length of a hull is getting a form**

$$\begin{aligned} q(x) &= -m(x)g + \rho g A(x) \\ &-m(x)\left(\ddot{\eta}_3 - x\ddot{\eta}_5\right) + F(x) \end{aligned}$$



BM & SF at a section x_p

- Total vertical shear force and bending moment at section x_p can be obtained by integrating load/ship length along ship length from the stern up to the section x'_p as follows

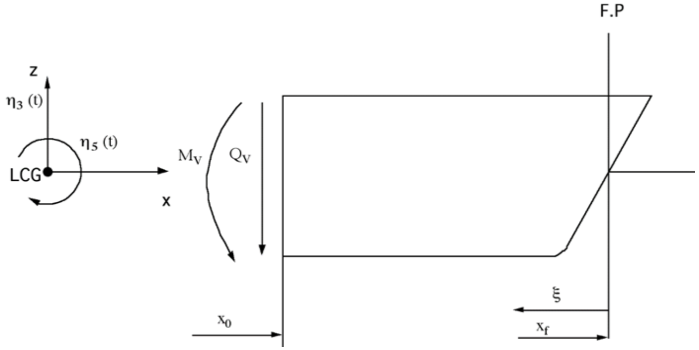
$$Q(x'_p) = \square_{0}^{x'_p} q(x')dx' \qquad M(x'_p) = \square_{0}^{x'_p} x'q(x')dx'$$

- The shear force and the bending moment are zero at the bow and at the stern

- If we subtract from the above expressions the still water values of shear force and bending moment respectively we get a linear approximation of the internal load distribution along the ship length in relation to wave actions.

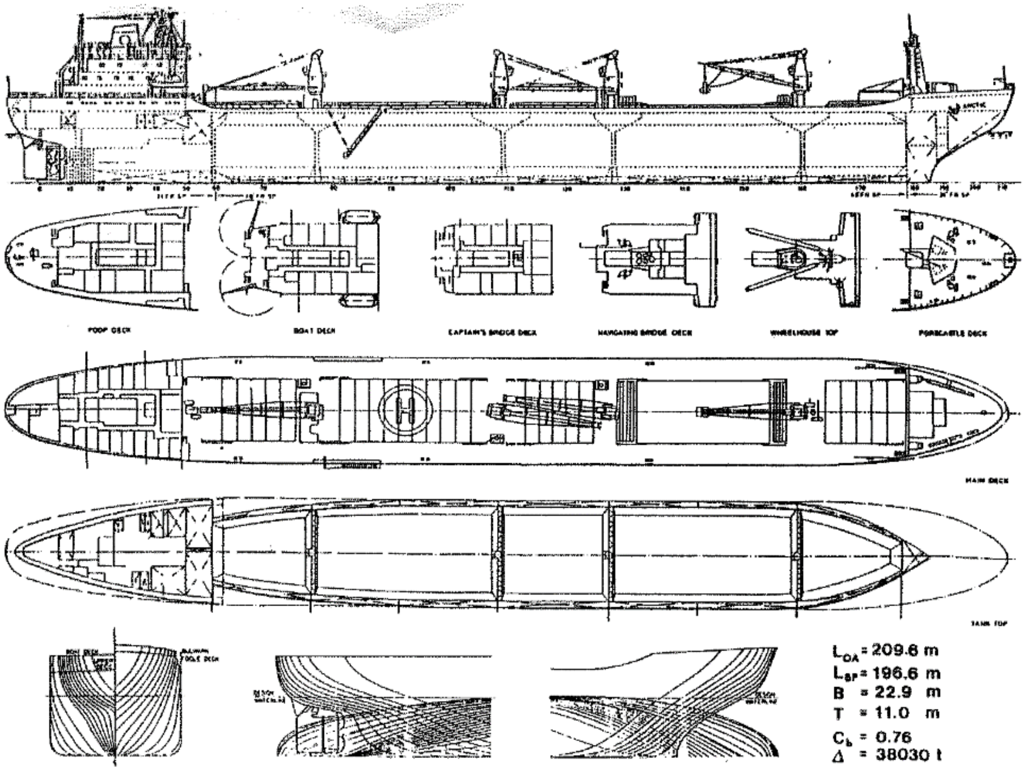
$$Q_V = e^{-i\omega t}\int_{x_0}^{x_f}(\eta_3 - x\eta_5)\left(f_3 - \omega^2 m(x) + f_{w3}\right)dx$$

$$M_V = -\int_{0}^{x_f - x_0} Q_V\,dx$$

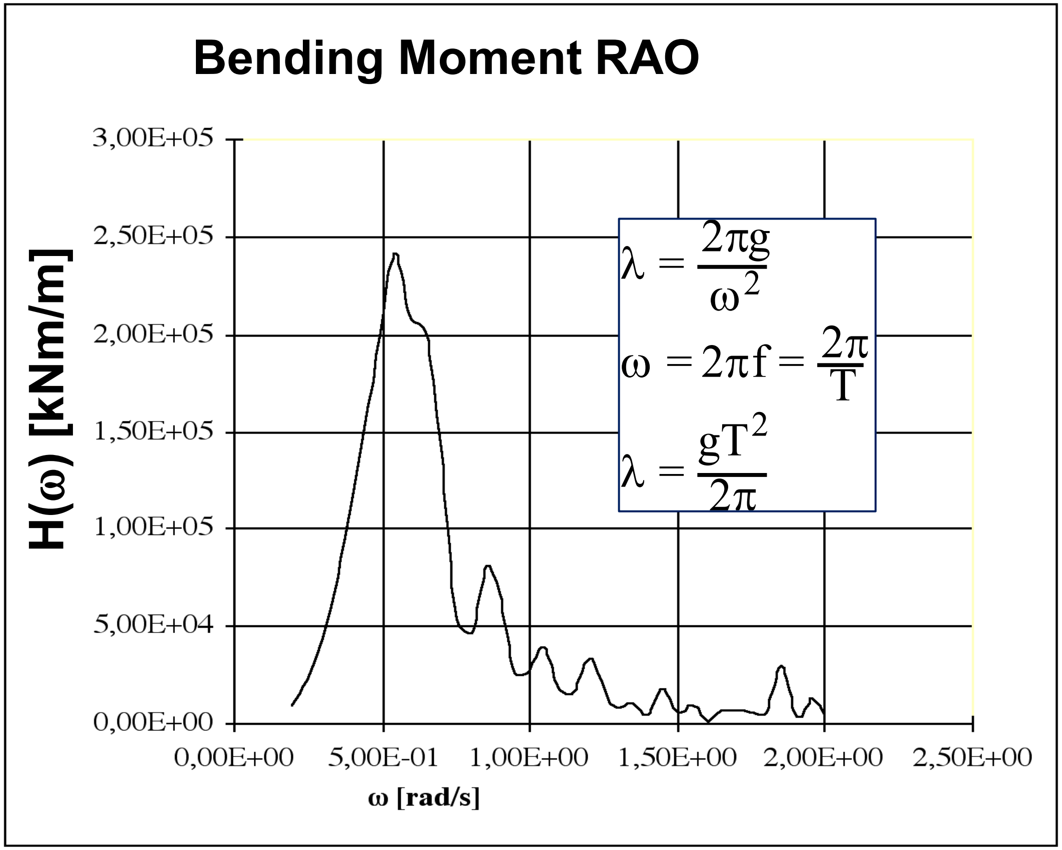


Example – MV Arctic

- L_{BP} = 196,59 m
- B = 22,86 m
- T = 10,97 m
- C_B = 0,76
- Δ = 38.030 ton
- DW = 28.000 ton
- Allowed M_{SW} 924,5 MNm (92.450 tonm)
- Section modulus:
  - Z_{deck} 12,982 m^3
  - Z_{bottom} 14,627 m^3
- Lloyd's 100 A1, Ice Class AC 2
- NS Steel R_e = 235 N/mm^2



MV Arctic (F_n= 0,17, χ=180°)



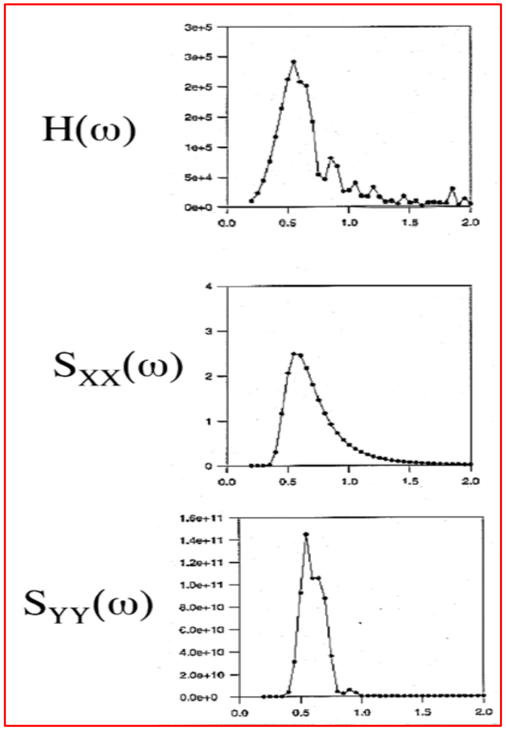
$$\lambda = \frac{2\pi g}{\omega^2}$$

$$\omega = 2\pi f = \frac{2\pi}{T}$$

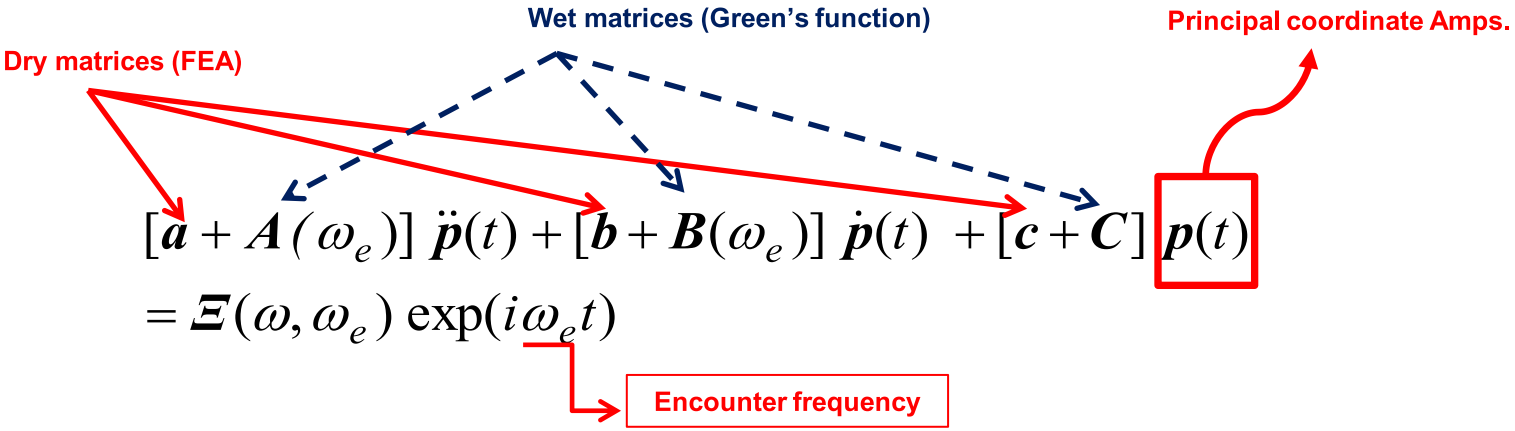
$$\lambda = \frac{gT^2}{2\pi}$$

- Transfer function $H(\omega)$ [kNm/m]
- Wave spectrum [m^2s]
  - T_z = 8.5 s
  - H_s = 4.0 m
- ***Response spectrum***

$S_{YY}(\omega) = H^2(\omega)\, S_{XX}(\omega)[(kNm/m)^2m^2s]$



Hydroelasticity of Ships – brief Introduction



Wet matrices (Green's function)

Principal coordinate Amps.

Dry matrices (FEA)

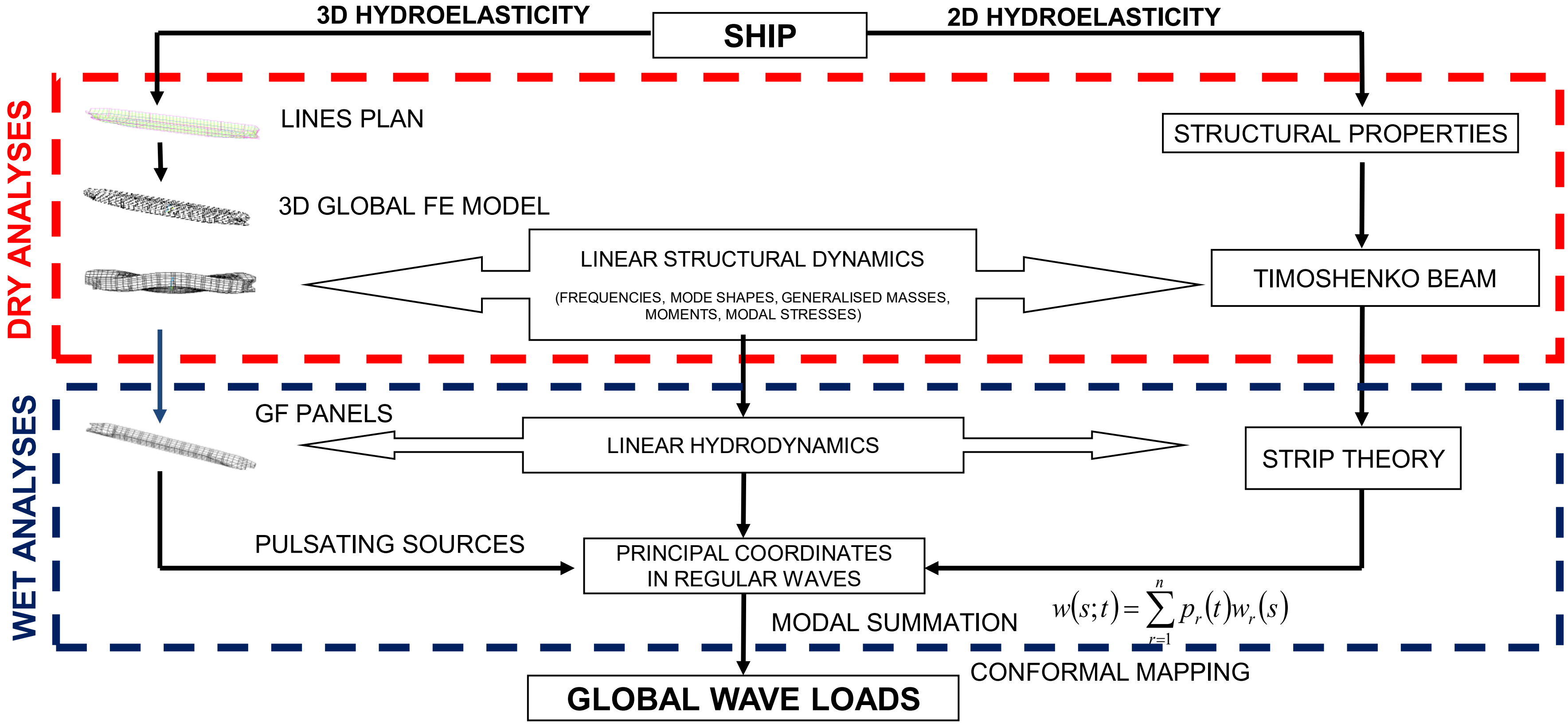
$$[\boldsymbol{a}+\boldsymbol{A}(\omega_e)]\,\ddot{\boldsymbol{p}}(t)+[\boldsymbol{b}+\boldsymbol{B}(\omega_e)]\,\dot{\boldsymbol{p}}(t)+[\boldsymbol{c}+\boldsymbol{C}]\,\boxed{\boldsymbol{p}(t)}$$
$$=\Xi(\omega,\omega_e)\exp(i\omega_e t)$$

Encounter frequency

Unified Hydroelasticity	**DRY analysis**	**WET analysis**
2D	Beam theory (Analytical, FD, FEA)	Strip theory (conformal mapping)
3D	3D FEA (shell, beam elements)	Green function (pulsating source)

Hirdaris, S.E., Price, W.G. and Temarel, P.: Two- and Three-dimensional Hydroelastic Analysis of a Bulker in Waves. *Marine Structures* ,16:627-65.

Hydroelasticity of Ships – brief Introduction



3D HYDROELASTICITY

SHIP

2D HYDROELASTICITY

DRY ANALYSES

LINES PLAN

3D GLOBAL FE MODEL

STRUCTURAL PROPERTIES

LINEAR STRUCTURAL DYNAMICS

(FREQUENCIES, MODE SHAPES, GENERALISED MASSES, MOMENTS, MODAL STRESSES)

TIMOSHENKO BEAM

WET ANALYSES

GF PANELS

LINEAR HYDRODYNAMICS

STRIP THEORY

PULSATING SOURCES

PRINCIPAL COORDINATES IN REGULAR WAVES

MODAL SUMMATION

$$w(s;t)=\sum_{r=1}^{n} p_r(t)\,w_r(s)$$

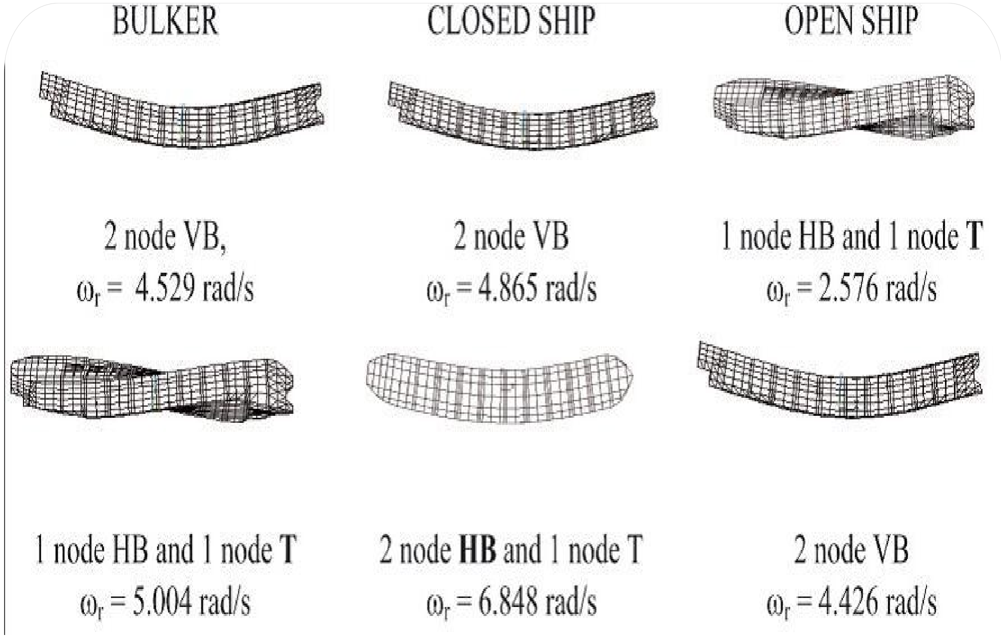
CONFORMAL MAPPING

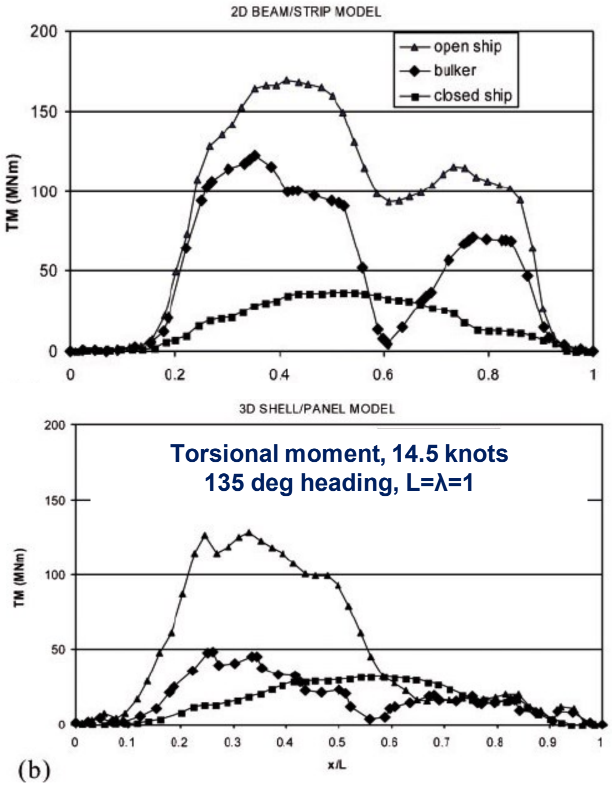
GLOBAL WAVE LOADS

Paper : Hirdaris, S.E., Price, W.G. and Temarel, P. (2003) Two- and Three-dimensional Hydroelastic Analysis of a Bulker in Waves. *Marine Structures,* 16:627-65

Hydroelasticity of Ships – brief Introduction

Discontinuities on dry hull modes- Based on inclusion or not of deck strips



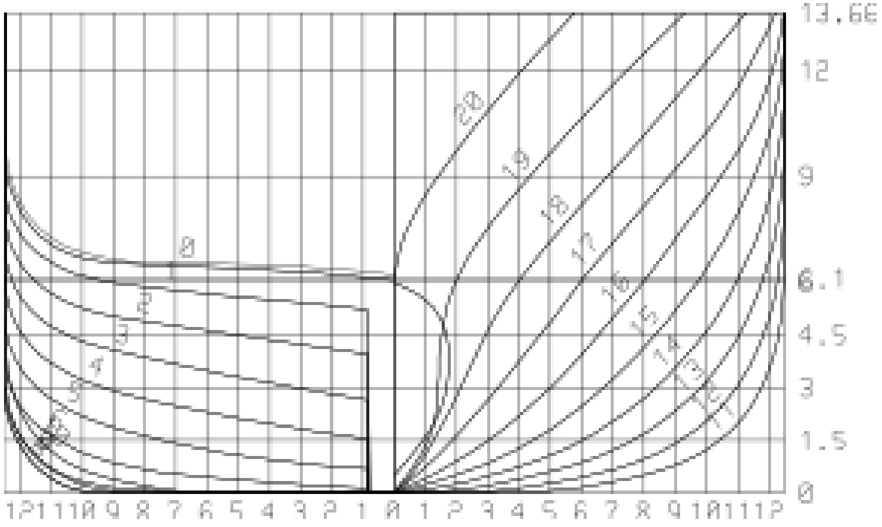


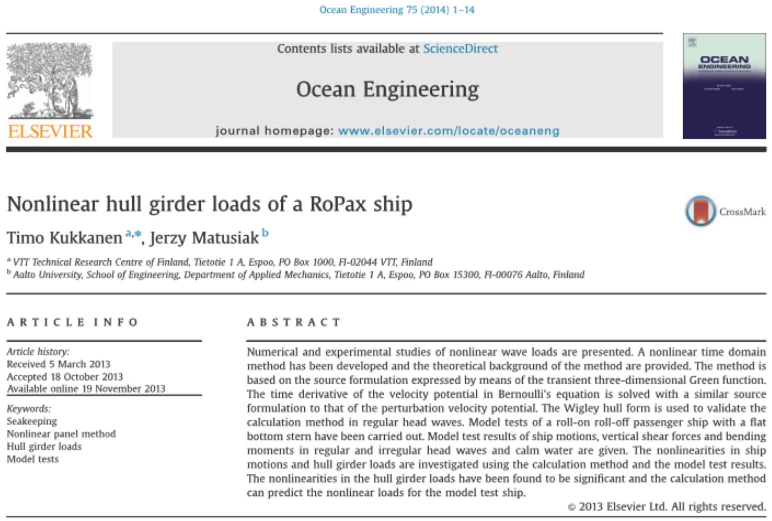
Torsional moment, 14.5 knots
135 deg heading, L=λ=1

Paper (2009) : Hirdaris, S.E. and Temarel, P. Hydroelasticity of Ships – recent advances and future trends. *Proc. IMechE, Part M: J. of Eng. Mar. Env.*, 223(3):305-330.

Example of Non-Linear Case (Ro-Pax Ship)

Quantity	Symbol	Unit	Value
Length over all	L_{oa}	[m]	171.4
Length between perpendiculars	L_{pp}	[m]	158.0
Breadth max. at waterline	B_{wl}	[m]	25.0
Draught	T	[m]	6.1
Displacement	∇	[m^3]	13 766
Block coefficient	C_B	–	0.55
Centre of gravity:			
From AP	x_{CG}	[m]	74.9
From CL	y_{CG}	[m]	0.0
From BL	z_{CG}	[m]	10.9
Radius of gyration in pitch	k_{yy}/L_{pp}	–	0.25
Transverse metacentric height	GM_{T0}	[m]	2.8





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Nonlinear hull girder loads of a RoPax ship

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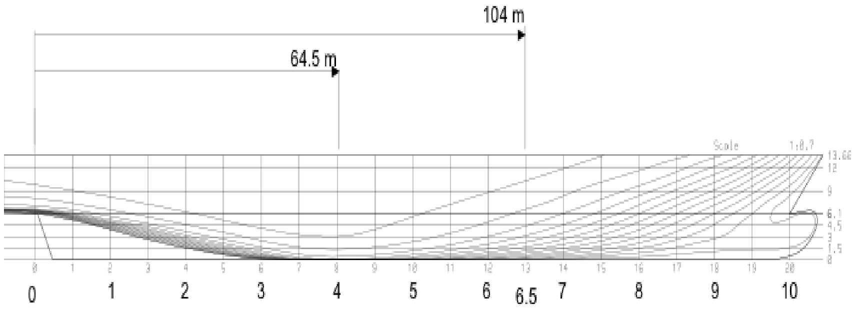
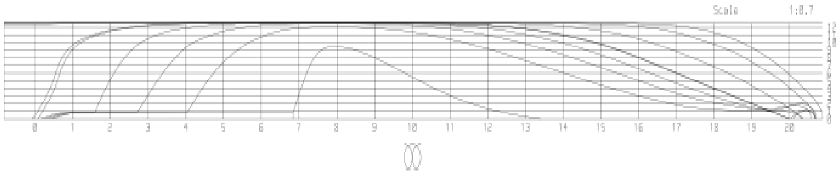
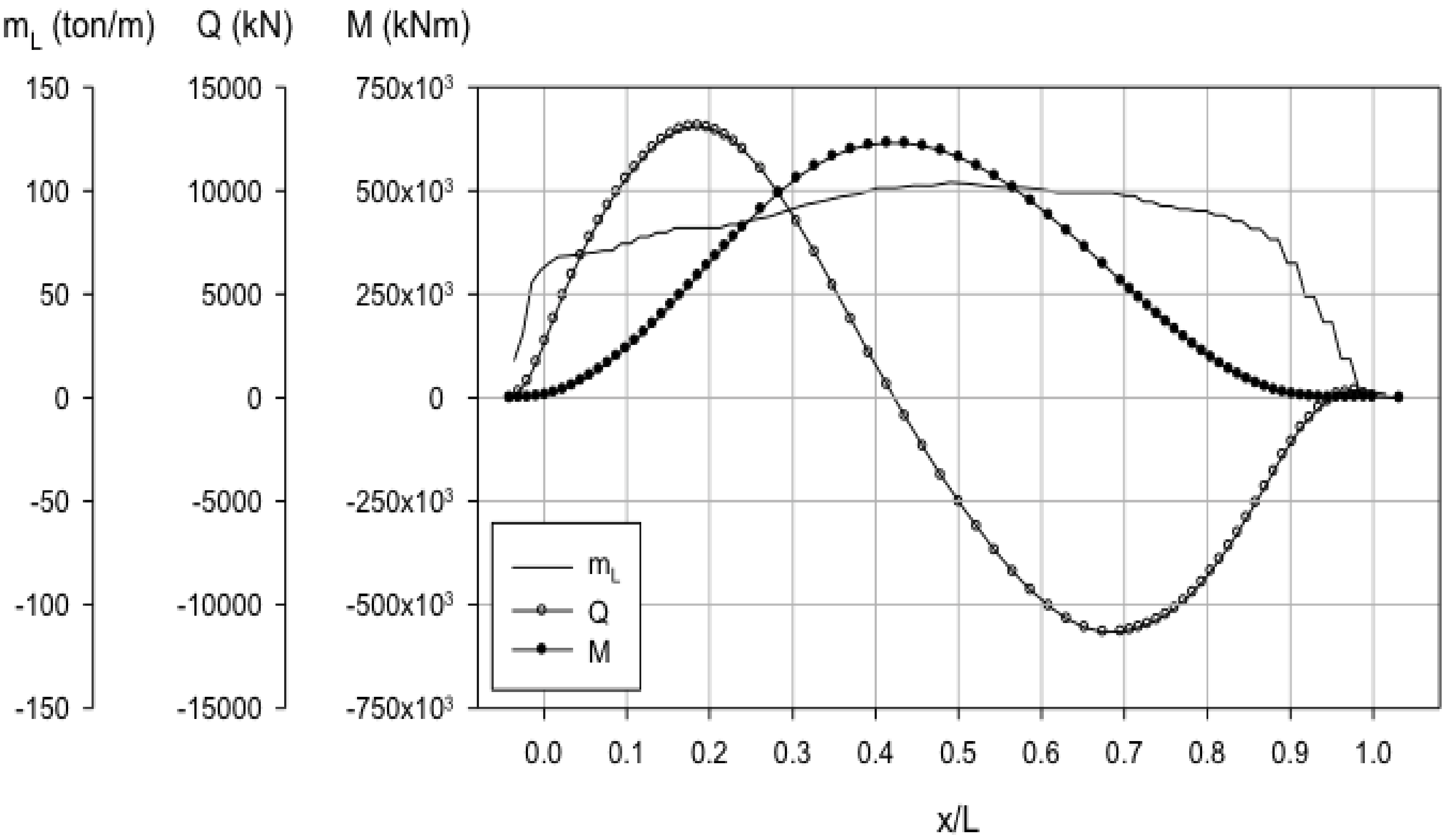
ABSTRACT

Numerical and experimental studies of nonlinear wave loads are presented. A nonlinear time domain method has been developed and the theoretical background of the method are provided. The method is based on the source formulation expressed by means of the transient three-dimensional Green function. The time derivative of the velocity potential in Bernoulli's equation is solved with a similar source formulation to that of the perturbation velocity potential. The Wigley hull form is used to validate the calculation method in regular head waves. Model tests of a roll-on roll-off passenger ship with a flat bottom stern have been carried out. Model test results of ship motions, vertical shear forces and bending moments in regular and irregular head waves and calm water are given. The nonlinearities in ship motions and hull girder loads are investigated using the calculation method and the model test results. The nonlinearities in the hull girder loads have been found to be significant and the calculation method can predict the nonlinear loads for the model test ship.

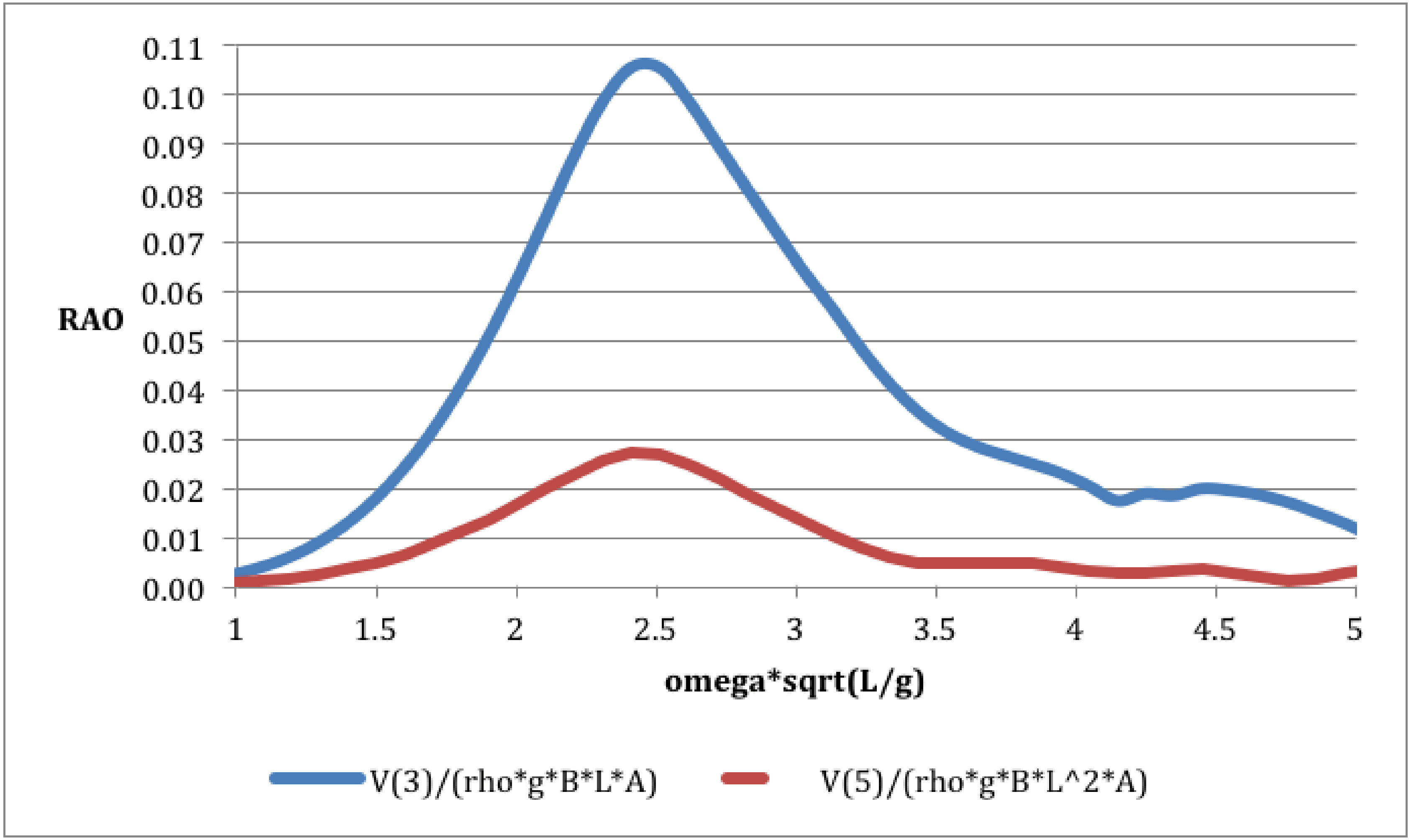




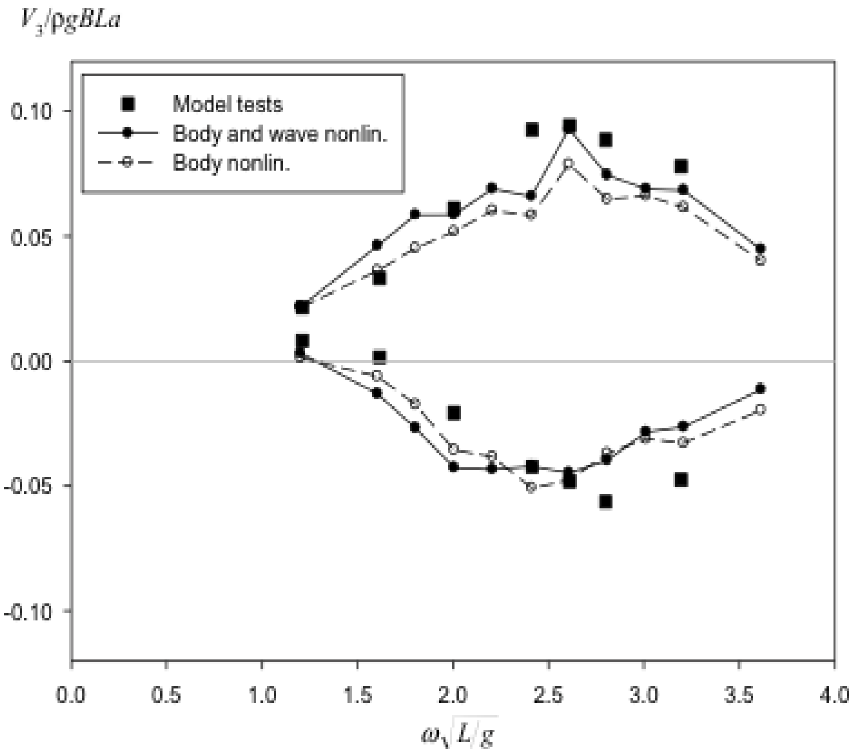
Weight, VSF & VBM distributions



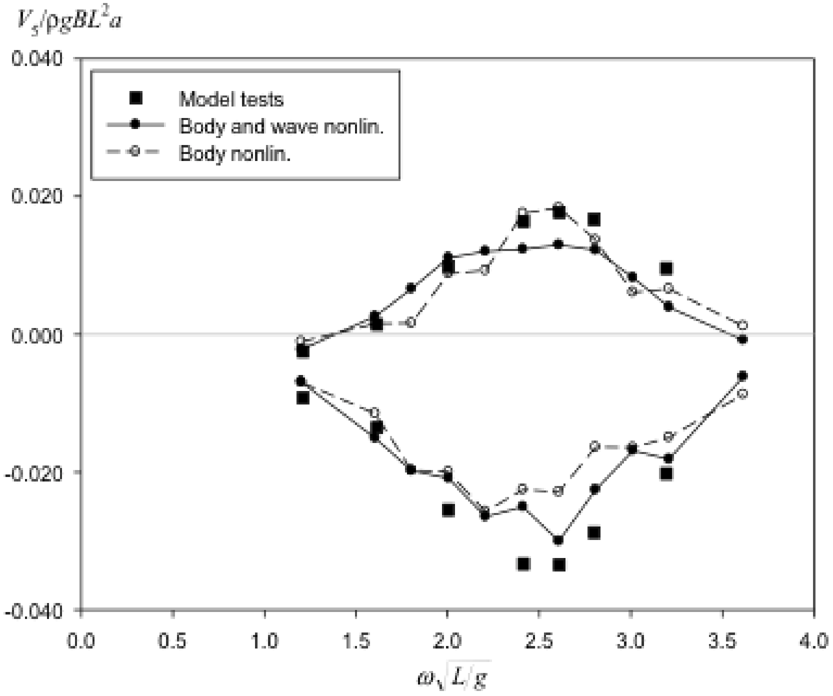
RAO of VSF & VBM - (χ = 180, Fn = 0.25)



Sagging & Hogging (χ = 180, Fn = 0.25)



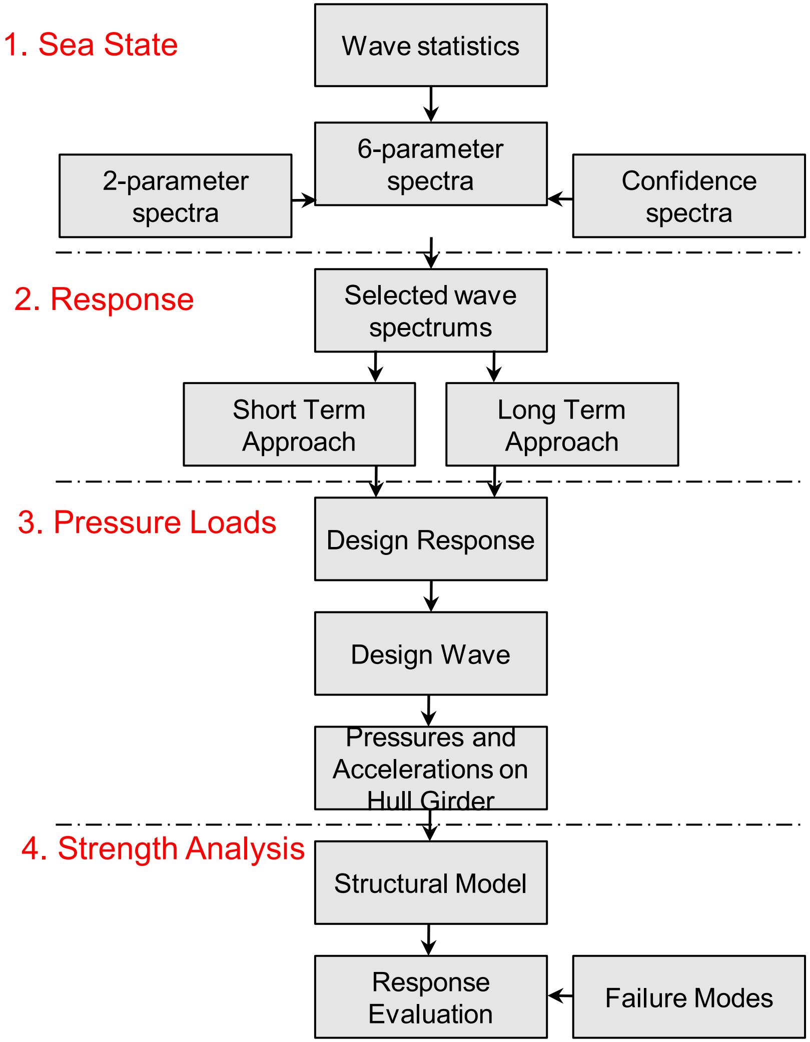
Significant differences Due to Non Linear Effects



Asymmetry

Design for Lifetime Service

- The basis is the description of sea state with wave spectrum
  - Years of wave measurements and resulting statistics such as H_S, T_Z, (BMT wave stats)
  - Wave spectrums
- Two things are of interest
  - Short term response *(M, Q)*
  - Long term response *(M, Q)*
- Short term response is used when ultimate strength is considered, i.e. strength against extreme loads
- Long term response is used when fatigue strength is considered, i.e. cumulative damage from years of operation – can be used also to predict the ultimate loads (pay attention to extrapolation from short term maxima)



1. Sea State
Wave statistics
2-parameter spectra
6-parameter spectra
Confidence spectra

2. Response
Selected wave spectrums
Short Term Approach
Long Term Approach

3. Pressure Loads
Design Response
Design Wave
Pressures and Accelerations on Hull Girder

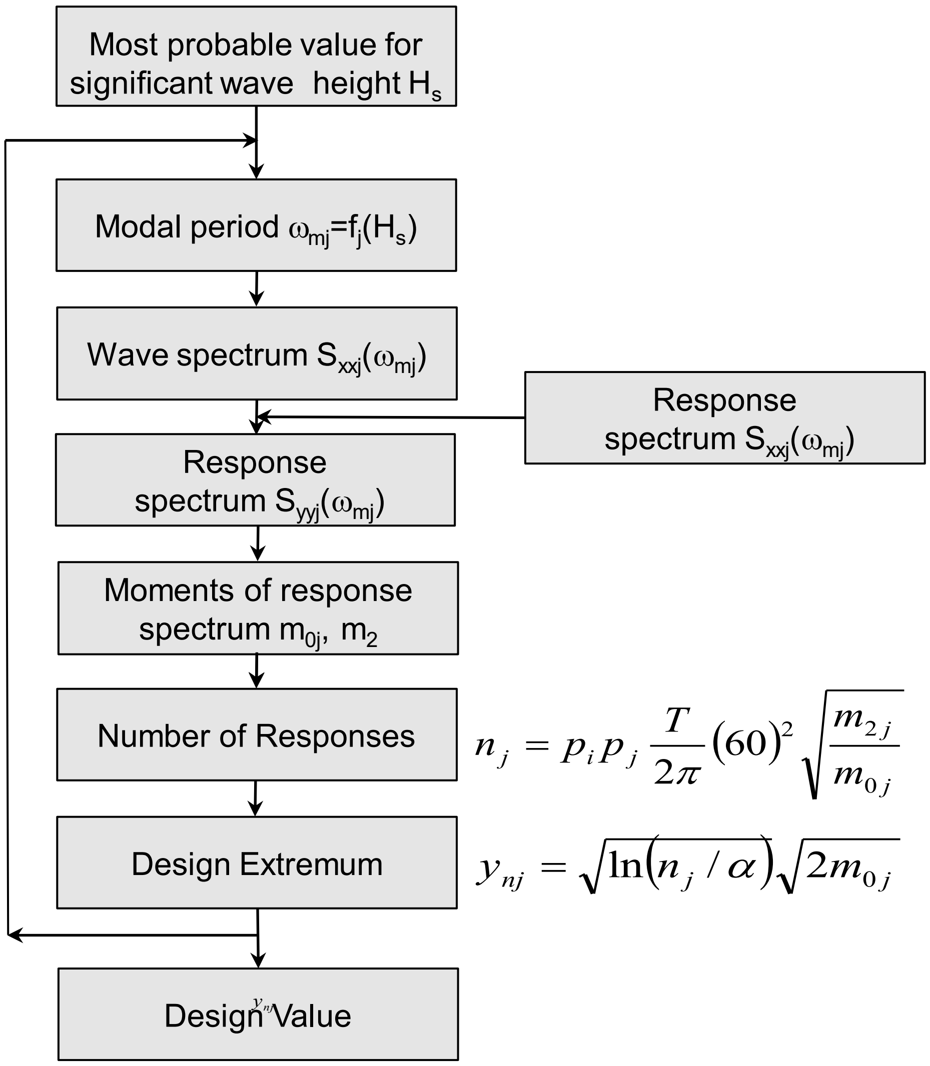
4. Strength Analysis
Structural Model
Response Evaluation
Failure Modes

ST Response from Extreme Sea State

- The aim is to find the response under extreme load conditions, i.e. reserve for ultimate strength

- The interesting failure modes are those which can happen during one load cycle:
  - Rupture
  - Buckling

- The idea is to find the extreme value for wave height H_s
  - Wave statistics
  - Entire lifetime, 20-25 year for ships, 100 years for offshore
  - 20 years is T =20*365*24h.



Most probable value for significant wave height H_s

Modal period $\omega_{mj}=f_j(H_s)$

Wave spectrum $S_{xxj}(\omega_{mj})$

Response spectrum $S_{xxj}(\omega_{mj})$

Response spectrum $S_{yyj}(\omega_{mj})$

Moments of response spectrum m_{0j}, m_2

Number of Responses

$$n_j = p_i p_j \frac{T}{2\pi}(60)^2\sqrt{\frac{m_{2j}}{m_{0j}}}$$

Design Extremum

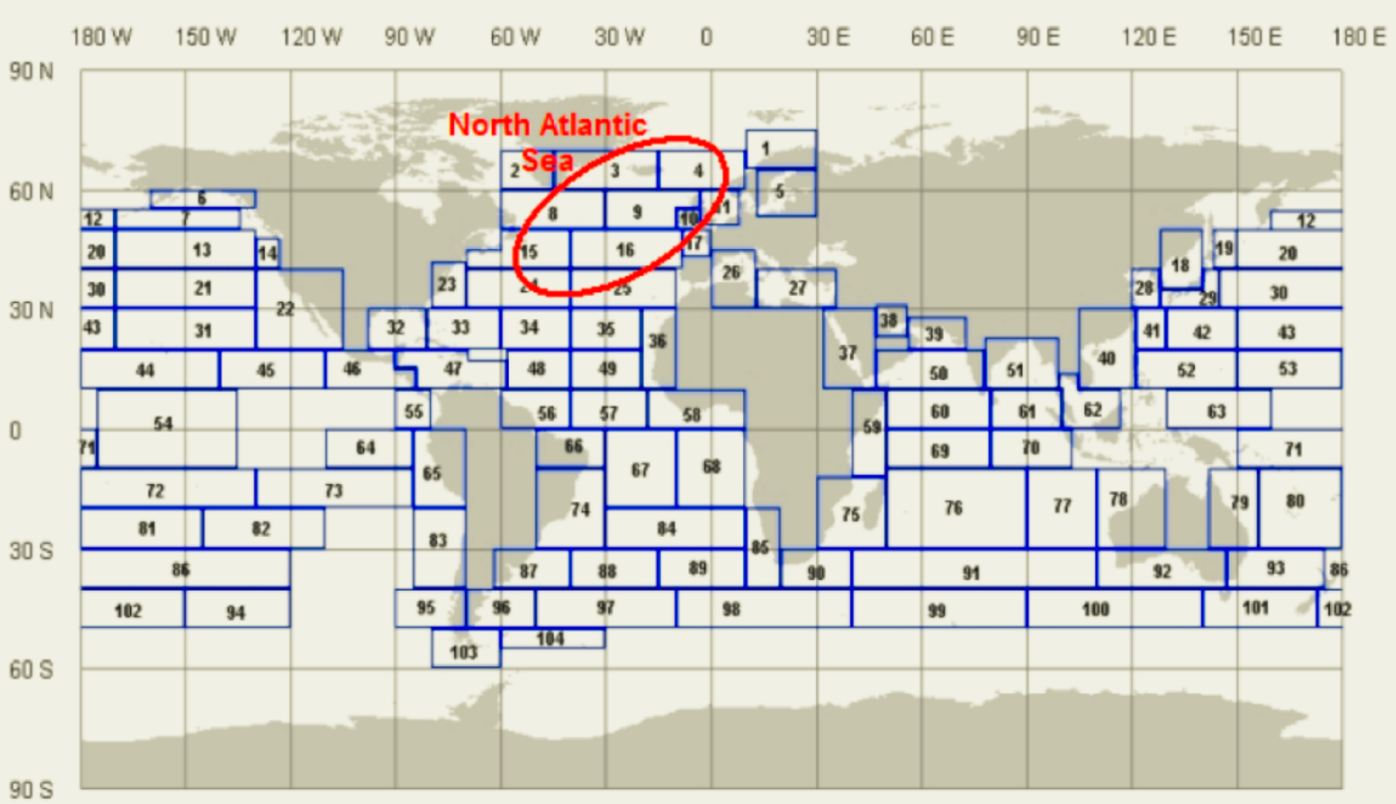
$$y_{nj} = \sqrt{\ln(n_j/\alpha)}\sqrt{2m_{0j}}$$

Design y_{nj} Value

Assumed Sea State

The North-Atlantic is usually used in design of ship structures :

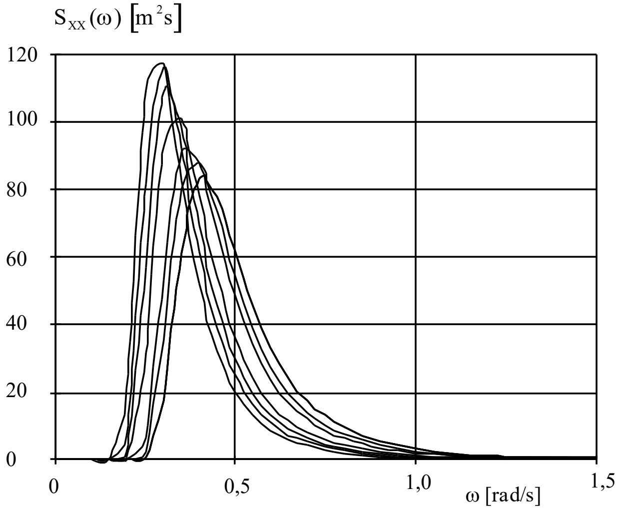
- If unlimited operation area is considered
- It is not wise to consider this always – too high loads?



Family of Wave Spectrums

- Many standard spectrum exists
  - modified Pierson-Moskowitz
  - Bretschneider
  - ITTC
  - ISSC
  - JONSWAP

- Full developed sea state is described by 2-parameter Pierson-Moskowitz wave spectrum
  - Duration is large
  - Fetch is large

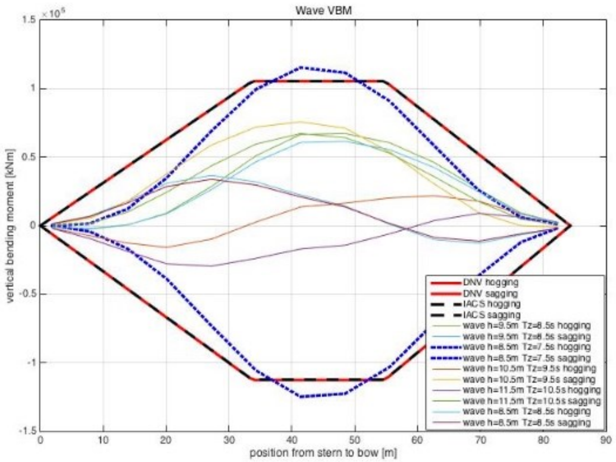


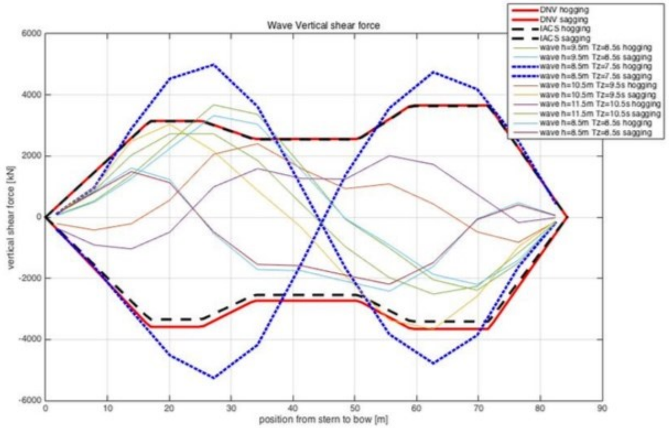
No.	ω_m [rad/s] H_S = 19,6 m
1	0,277
2	0,295
3	0,311
4	0,331
5	0,368
6	0,386
7	0,410
8	0,437
9	0,460

$$S_{xx}(\omega) = \frac{1}{4\pi} H_s^2 \frac{1}{1.41\omega_m} \left(\frac{\omega}{1.41\omega_m} \right)^{-5} \cdot e^{-\frac{1}{\pi}\left(\frac{\omega}{1.41\omega_m}\right)^{-4}} \left[\frac{m^2}{s} \right]$$

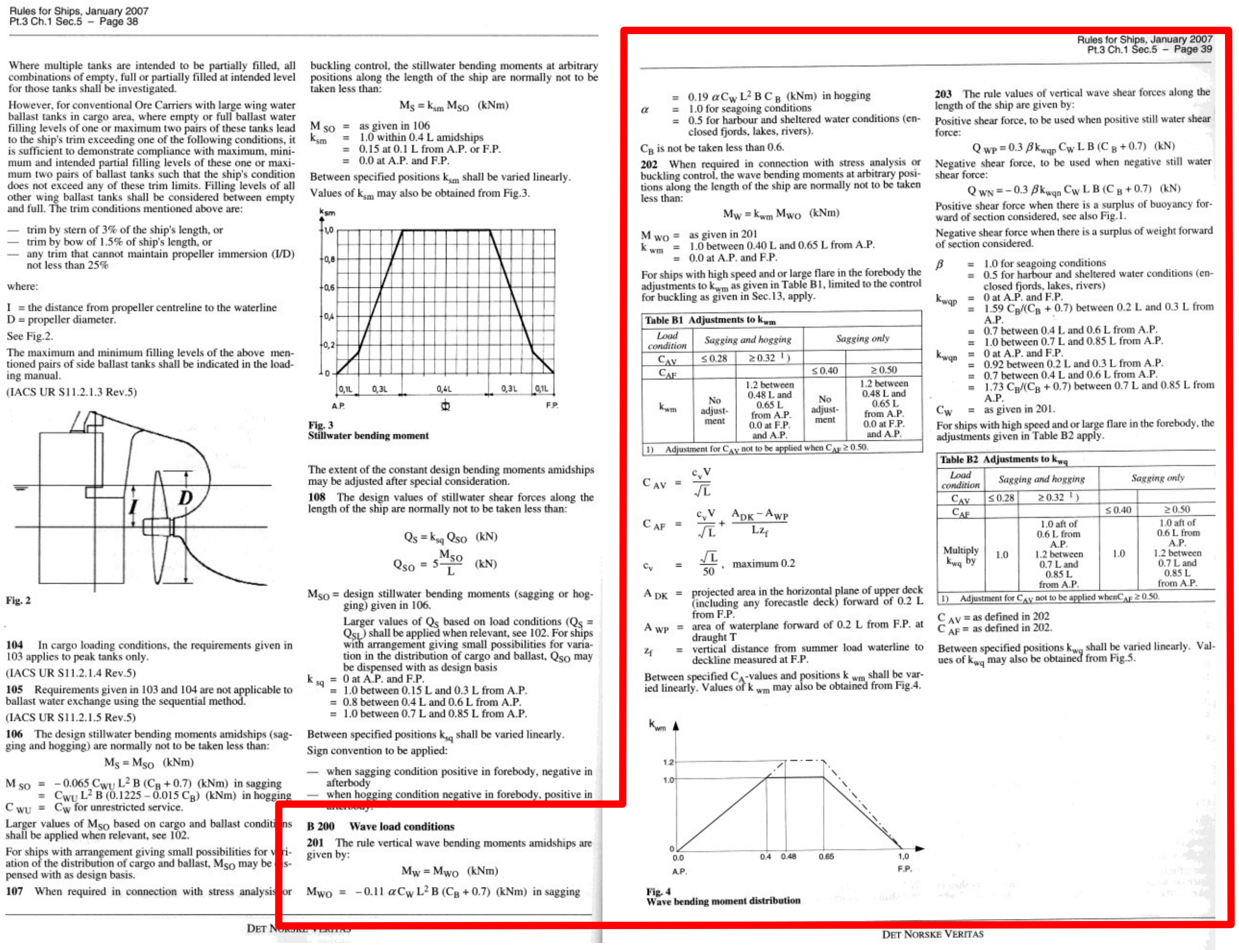
Hull Girder Global Loads in Rules

- The simulations lead results for one sea state
- Over the lifetime several load conditions occur and all of them must be checked
- The rules use “envelope curves” for the longitudinal moment and shear force distributions
- These are not fulfilling the basic rules of statics $q=dQ/dx=d^2M/dx^2$
- The envelope curves are easy to apply (e.g. FEM) and correct for non-linearity





Classification Society Bending Moment (Simplified)
Short Term Response (Buckling and Yielding)



Where multiple tanks are intended to be partially filled, all combinations of empty, full or partially filled at intended level for those tanks shall be investigated.

However, for conventional Ore Carriers with large wing water ballast tanks in cargo area, where empty or full ballast water filling levels of one or maximum two pairs of these tanks lead to the ship's trim exceeding one of the following conditions, it is sufficient to demonstrate compliance with maximum, minimum and intended partial filling levels of these one or maximum two pairs of ballast tanks such that the ship's condition does not exceed any of these trim limits. Filling levels of all other wing ballast tanks shall be considered between empty and full. The trim conditions mentioned above are:

- trim by stern of 3% of the ship's length, or
- trim by bow of 1.5% of ship's length, or
- any trim that cannot maintain propeller immersion (I/D) not less than 25%

where:

I = the distance from propeller centreline to the waterline
D = propeller diameter.

See Fig.2.

The maximum and minimum filling levels of the above mentioned pairs of side ballast tanks shall be indicated in the loading manual.

(IACS UR S11.2.1.3 Rev.5)

Fig. 2

104 In cargo loading conditions, the requirements given in 103 applies to peak tanks only.

(IACS UR S11.2.1.4 Rev.5)

105 Requirements given in 103 and 104 are not applicable to ballast water exchange using the sequential method.

(IACS UR S11.2.1.5 Rev.5)

106 The design stillwater bending moments amidships (sagging and hogging) are normally not to be taken less than:

$$M_S = M_{SO} \quad (\text{kNm})$$

M_{SO} = $-0.065\, C_{WU} L^2 B (C_B + 0.7)$ (kNm) in sagging
= $C_{WU} L^2 B (0.1225 - 0.015\, C_B)$ (kNm) in hogging
C_{WU} = C_W for unrestricted service.

Larger values of M_{SO} based on cargo and ballast conditions shall be applied when relevant, see 102.

For ships with arrangement giving small possibilities for variation of the distribution of cargo and ballast, M_{SO} may be dispensed with as design basis.

107 When required in connection with stress analysis or buckling control, the stillwater bending moments at arbitrary positions along the length of the ship are normally not to be taken less than:

$$M_S = k_{sm} M_{SO} \quad (\text{kNm})$$

M_{SO} = as given in 106
k_{sm} = 1.0 within 0.4 L amidships
= 0.15 at 0.1 L from A.P. or F.P.
= 0.0 at A.P. and F.P.

Between specified positions k_{sm} shall be varied linearly.

Values of k_{sm} may also be obtained from Fig.3.

Fig. 3
Stillwater bending moment

The extent of the constant design bending moments amidships may be adjusted after special consideration.

108 The design values of stillwater shear forces along the length of the ship are normally not to be taken less than:

$$Q_S = k_{sq} Q_{SO} \quad (\text{kN})$$

$$Q_{SO} = 5\frac{M_{SO}}{L} \quad (\text{kN})$$

M_{SO} = design stillwater bending moments (sagging or hogging) given in 106.

Larger values of Q_S based on load conditions ($Q_S = Q_{SL}$) shall be applied when relevant, see 102. For ships with arrangement giving small possibilities for variation in the distribution of cargo and ballast, Q_{SO} may be dispensed with as design basis

k_{sq} = 0 at A.P. and F.P.
= 1.0 between 0.15 L and 0.3 L from A.P.
= 0.8 between 0.4 L and 0.6 L from A.P.
= 1.0 between 0.7 L and 0.85 L from A.P.

Between specified positions k_{sq} shall be varied linearly.

Sign convention to be applied:

- when sagging condition positive in forebody, negative in afterbody
- when hogging condition negative in forebody, positive in afterbody.

B 200 Wave load conditions

201 The rule vertical wave bending moments amidships are given by:

$$M_W = M_{WO} \quad (\text{kNm})$$

M_{WO} = $-0.11\, \alpha C_W L^2 B (C_B + 0.7)$ (kNm) in sagging

= $0.19\, \alpha C_W L^2 B C_B$ (kNm) in hogging
α = 1.0 for seagoing conditions
= 0.5 for harbour and sheltered water conditions (enclosed fjords, lakes, rivers).

C_B is not be taken less than 0.6.

202 When required in connection with stress analysis or buckling control, the wave bending moments at arbitrary positions along the length of the ship are normally not to be taken less than:

$$M_W = k_{wm} M_{WO} \quad (\text{kNm})$$

M_{WO} = as given in 201
k_{wm} = 1.0 between 0.40 L and 0.65 L from A.P.
= 0.0 at A.P. and F.P.

For ships with high speed and or large flare in the forebody the adjustments to k_{wm} as given in Table B1, limited to the control for buckling as given in Sec.13, apply.

Table B1 Adjustments to k_{wm}

Load condition	*Sagging and hogging*		*Sagging only*	
C_{AV}	≤ 0.28	≥ 0.32 1)		
C_{AF}			≤ 0.40	≥ 0.50
k_{wm}	No adjustment	1.2 between 0.48 L and 0.65 L from A.P. 0.0 at F.P. and A.P.	No adjustment	1.2 between 0.48 L and 0.65 L from A.P. 0.0 at F.P. and A.P.

1) Adjustment for C_{AV} not to be applied when $C_{AF} \geq 0.50$.

$$C_{AV} = \frac{c_v V}{\sqrt{L}}$$

$$C_{AF} = \frac{c_v V}{\sqrt{L}} + \frac{A_{DK} - A_{WP}}{L z_f}$$

$$c_v = \frac{\sqrt{L}}{50}, \quad \text{maximum } 0.2$$

A_{DK} = projected area in the horizontal plane of upper deck (including any forecastle deck) forward of 0.2 L from F.P.
A_{WP} = area of waterplane forward of 0.2 L from F.P. at draught T
z_f = vertical distance from summer load waterline to deckline measured at F.P.

Between specified C_A-values and positions k_{wm} shall be varied linearly. Values of k_{wm} may also be obtained from Fig.4.

Fig. 4
Wave bending moment distribution

203 The rule values of vertical wave shear forces along the length of the ship are given by:

Positive shear force, to be used when positive still water shear force:

$$Q_{WP} = 0.3\, \beta k_{wqp} C_W L B (C_B + 0.7) \quad (\text{kN})$$

Negative shear force, to be used when negative still water shear force:

$$Q_{WN} = -0.3\, \beta k_{wqn} C_W L B (C_B + 0.7) \quad (\text{kN})$$

Positive shear force when there is a surplus of buoyancy forward of section considered, see also Fig.1.

Negative shear force when there is a surplus of weight forward of section considered.

β = 1.0 for seagoing conditions
= 0.5 for harbour and sheltered water conditions (enclosed fjords, lakes, rivers)
k_{wqp} = 0 at A.P. and F.P.
= $1.59\, C_B/(C_B + 0.7)$ between 0.2 L and 0.3 L from A.P.
= 0.7 between 0.4 L and 0.6 L from A.P.
= 1.0 between 0.7 L and 0.85 L from A.P.
k_{wqn} = 0 at A.P. and F.P.
= 0.92 between 0.2 L and 0.3 L from A.P.
= 0.7 between 0.4 L and 0.6 L from A.P.
= $1.73\, C_B/(C_B + 0.7)$ between 0.7 L and 0.85 L from A.P.
C_W = as given in 201.

For ships with high speed and or large flare in the forebody, the adjustments given in Table B2 apply.

Table B2 Adjustments to k_{wq}

Load condition	*Sagging and hogging*		*Sagging only*	
C_{AV}	≤ 0.28	≥ 0.32 1)		
C_{AF}			≤ 0.40	≥ 0.50
Multiply k_{wq} by	1.0	1.0 aft of 0.6 L from A.P. 1.2 between 0.7 L and 0.85 L from A.P.	1.0	1.0 aft of 0.6 L from A.P. 1.2 between 0.7 L and 0.85 L from A.P.

1) Adjustment for C_{AV} not to be applied when $C_{AF} \geq 0.50$.

C_{AV} = as defined in 202
C_{AF} = as defined in 202.

Between specified positions k_{wq} shall be varied linearly. Values of k_{wq} may also be obtained from Fig.5.

Exercise for home study : Find DNV Classification note
When we use Rules and When Direct Calcs?

Summary

- Direct analysis is needed when your ship is not suitable for application rules
  - Operational profile
  - Geometry

- Stochastic loads can be assessed using spectral methods
  - Input: wave spectrum (frequency domain)
  - RAO (linear operator): strip method, panel method, experiments
  - Output: response (frequency domain)

- When you know the spectrum, you can define the maximum loads (probability theory for stochastic processes)
  - Storm (3 hour maximum) worst-case-scenario for ultimate strength, input vs. RAO
  - Lifetime (25-50 years), cumulative loads for fatigue